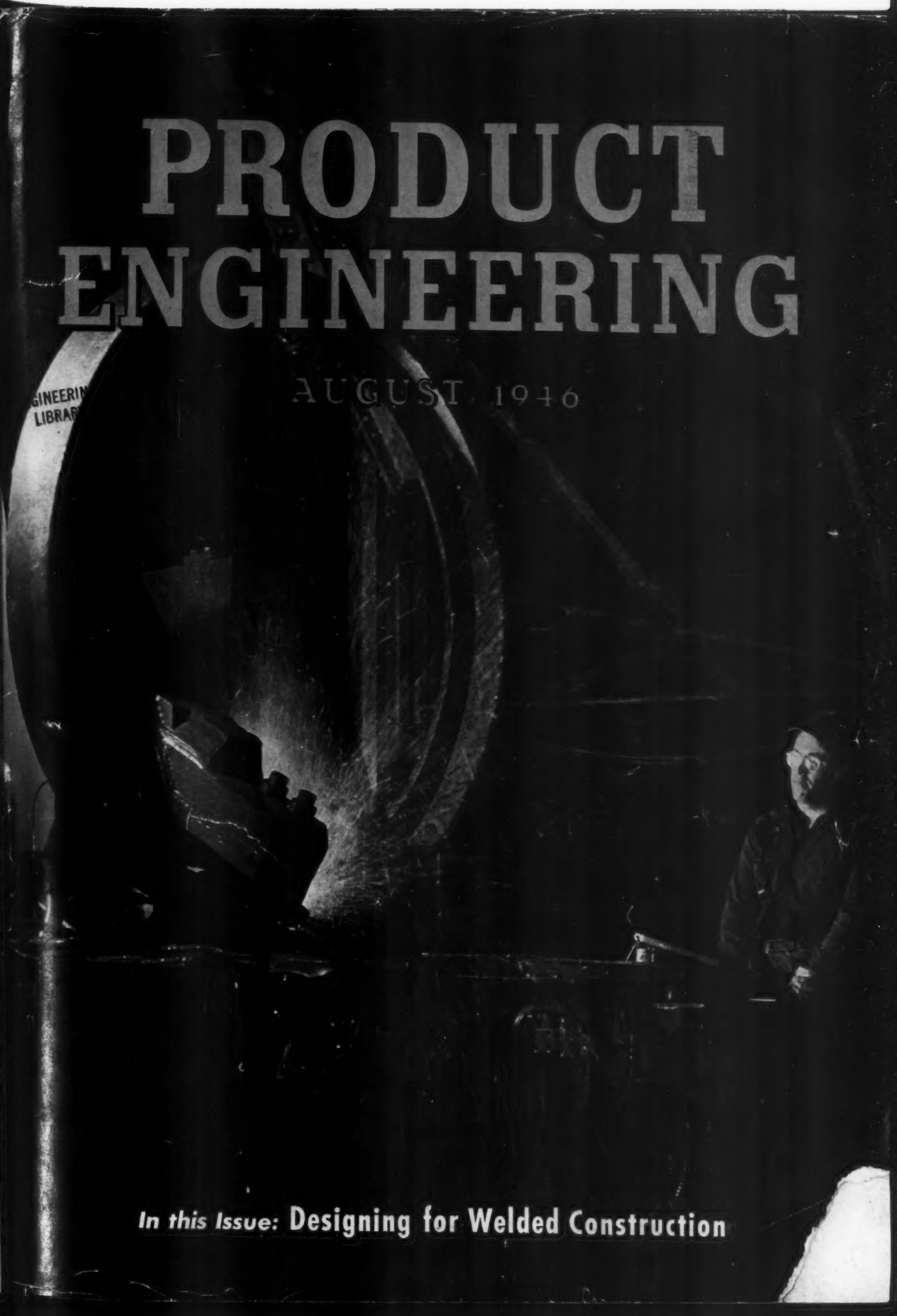
PRODUCT ENGINEERING

AUGUST 1946



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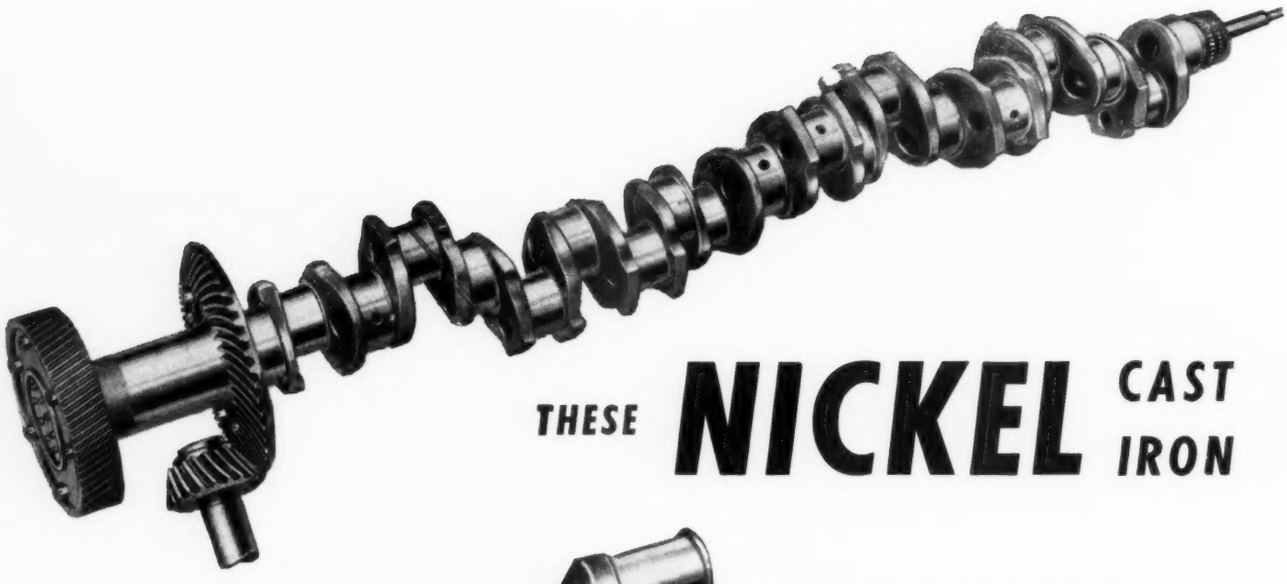

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★ **Excellent Vibration Damping**

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The smaller crank shown above is one of the types used in air compressors built by a national [illegible]*facturer.*

[illegible] *parts such as crankshafts where sharp changes of section are unavoidable, the low notch sensitivity of cast iron recommends its use. It also provides excellent vibration damping properties. Absence of expensive dies and forging equipment, and the fact that castings require less machine tool time . . . point to the economy of using Nickel alloy irons.*

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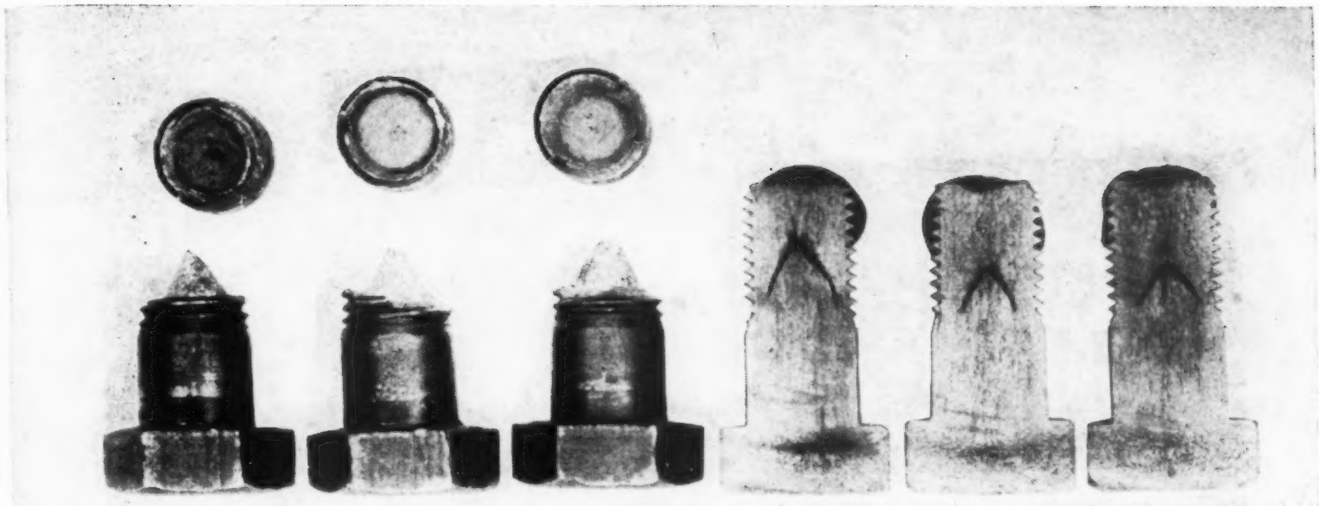


Fig. 1 — Internal static tension fractures resulting from excessive rolling of the material. The three bolts to the left failed at low loads. The internal fractures shown in the three longitudinal sections give some indication of the depth to which cold rolling affected the material.

Residual Stresses Can Be Beneficial Or Might Cause Failure of Part

J. O. ALMEN
Research Department, General Motors Corporation

Residual stresses may be an asset or a liability, depending on their relation to the stress pattern of the working load. Charts that show how stress raisers and stress corrosion affect the life of parts and curves that relate the life of parts to human life expectancy are included.

THE SIMPLEST of all design formulas is the fundamental relationship between tensile load, stress and cross-sectional area. Although few textbooks state it, and none explains it fully, the formula "Tensile load divided by cross-sectional area equals unit tensile stress" involves many assumptions. The formula gives the average unit stress, and then only if there are no residual stresses in the material. However, it is not the average stress but the maximum stress that causes a part in simple tension to fail. Hence the designer is not concerned with the average stress but rather the maximum stress that might be developed. The latter may be as much as several times the average.

Only when the element under tension has a smooth surface, uniform cross-section, concentric loading, and the material is homogeneous with reference to its physical properties and has been stress relieved throughout, is the average tensile stress nearly equal to the maximum. Such conditions are rarely found in machine elements. The following examples of the results of tests and investigations of failures of parts in service illustrate the degree to which ordinary manufacturing operations and processes can reduce the expected load carrying capacities of machine elements in simple tension when the effects of the processing operations are ignored and not taken into account.

Properly cold-rolled bolts are stronger in fatigue than are cut or ground bolts. To illustrate that good processes can lead to bad results, in Fig. 1 are shown six fractured bolts, three of them in longitudinal cross-section. The bolts have rolled threads and for the purpose of the test, they were rolled excessively. It is not to be assumed that these bolts are commercial articles; they were produced to demonstrate the ill effects that can result from excessive rolling or cold-working of the surface material without subsequent annealing.

The internal fractures shown in the longitudinally sectioned bolts were caused by the excessive rolling of the threads, the cold-working putting the surface layers of the material under high compression. The ends of the cracks indicate the approximate depth to which the material of the bolt was put under compression. Obviously, the total compressive force developed in the surface layers of the bolt had to be balanced by an equal tensile force in the core material, the average compression times the area over which it was distributed being equal to the average tension times the area over which it was distributed. The tension in the core

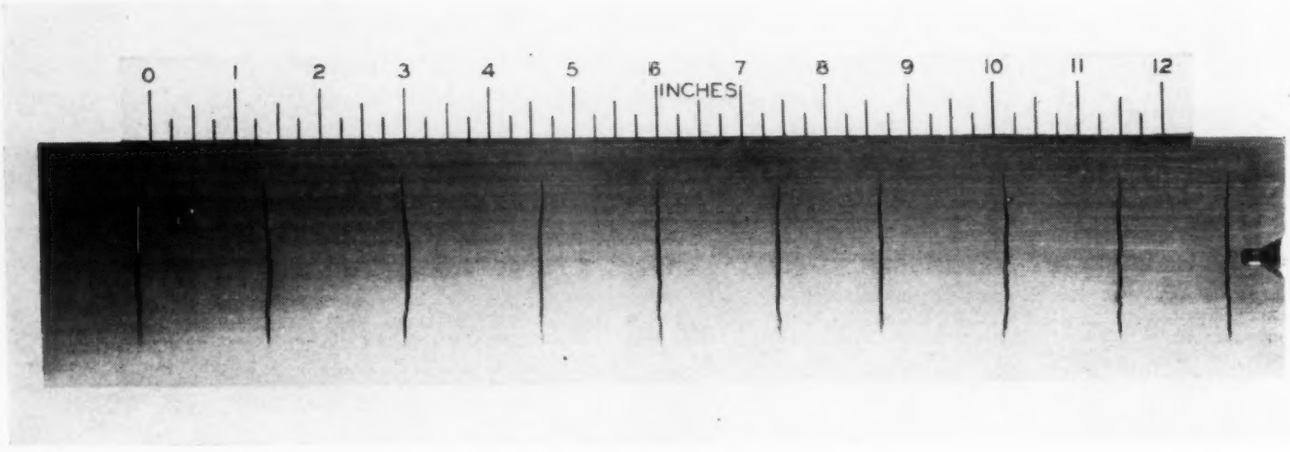
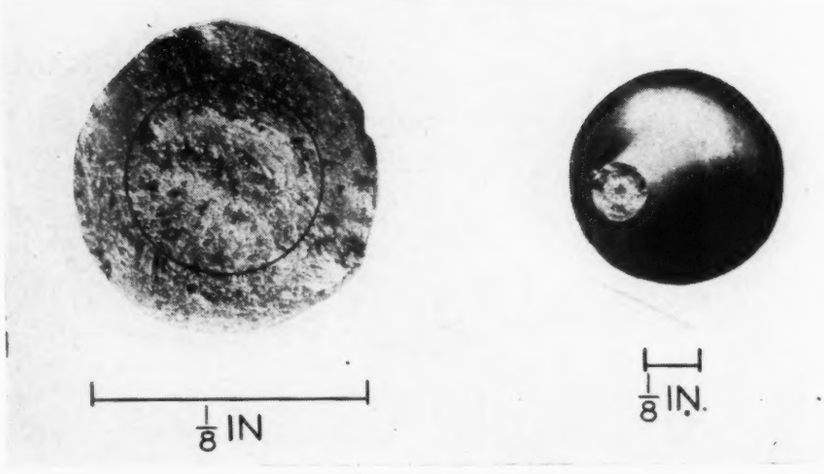
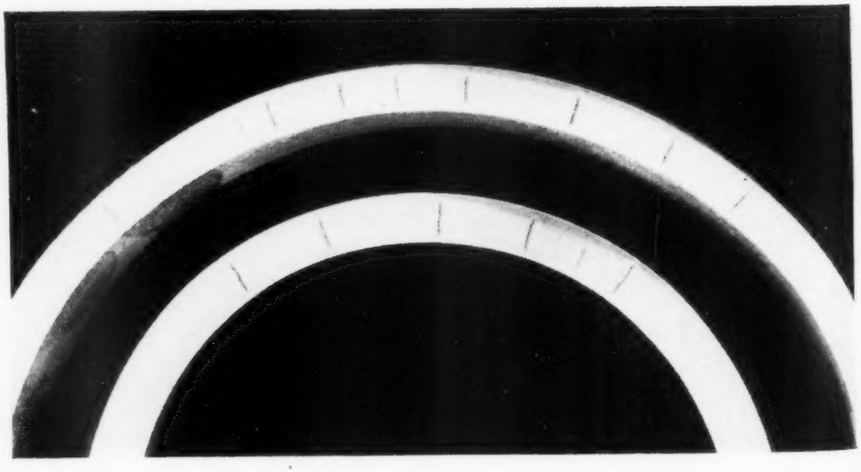


Fig. 2—Internal fractures in a bar that had been cold rolled excessively, thereby producing high tensile stresses in the core material.

Fig. 3—This failure was caused by severe tension stress resulting from deep nitriding developing compression stresses.

Fig. 4—Excessive internal tension stresses resulting from deep carburizing and severe quenching caused these thin rings to fracture internally.





material was so great that the material was ruptured even before any tension load was applied to the bolt, the internal rupture permitting the elongation of the bolt, thereby relieving the compressive stresses. But with a goodly portion of the cross-section of the bolt already ruptured, it failed upon the addition of a small applied load.

Cold-rolled bars that have been rolled excessively also show the same type of internal rupture caused for the same reason. Fig. 2 shows the cross-section of such a bar. The longitudinal distance between the transverse internal cracks is a measure of the relative depth and degree of cold-working of the given bar.

Tension Must Equal Compression

Because the compression in the surface layers must be balanced by tension in the core material, the greater the diameter of the core material relative to the cross-sectional area of material under compression, the lower will be the peak tensile stresses. The depth to which the material is cold-worked is relatively limited. Therefore, the larger the diameter of the bar, bolt, stud or other element, the lower will be the tensile stresses developed by cold-working. Conversely, thin sections are much more likely to fail as a result of compressive stresses in the surface layers.

Cold-rolling or cold-working is not the only process that produces compression in the surface of the material. In the nitriding process, the nitrides that are formed increase the mass or volume of the material. Therefore, the nitrided part as it comes from the furnace has the material in the nitrided surface layers in a state of compression. The stress condition is similar to that created by cold-working. In Fig. 3 is shown a machine part that had been nitrided. The high compressive stresses developed in the surface layers, by the increased mass of the nitrided surface material, created tension stresses of sufficient severity to cause fracture. The diameter of the stem was only $\frac{1}{8}$ in. A $\frac{1}{64}$ in. depth of nitrided case would have made the section areas under compression and tension almost equal. Hence the residual tension was almost equal to the compression developed by the swelling of the nitrided layers.

Excessive internal tension stresses can also result from the deep carburizing and rapid quenching of thin sections. Fig. 4 shows the etched cross-section of thin rings that failed from such cause. If the ring had been cooled sufficiently slowly after carburizing so

as to avoid the formation of martensite, the compressive surface stresses would not have developed and hence the internal fractures would have been avoided.

Cold-forming is another manufacturing operation that produces residual stresses. In the helical coil springs shown in Fig. 5, the cold-forming load was in a direction different from the subsequent working load, hence the residual stresses were harmful. The final stress distribution is appreciably different from the stresses as calculated in the conventional manner. In Fig. 5 is shown at (A), the "impact compression fractures" caused by residual tension. The diagram shows the actual and the calculated stresses. At (B) in Fig. 5 is shown the failure of a spring caused by residual tension developed during cold-forming. The distribution of the residual stresses across the section of the spring is shown by the diagram. In the conventional calculations these residual stresses are ignored.

Residual stresses can be harmful or beneficial, depending on how they superimpose on the stress pattern developed by the working loads. For example, because fatigue cracks always originate in surfaces that are under tension, not compression, the fatigue strength of a member can be increased by shot-peening the surfaces under tension. Thereby the peak tensile stresses at the surface are reduced and the fatigue life of the member correspondingly increased.

Shot-peened Leaf Springs

In Fig. 6 is shown diagrammatically how a leaf spring is greatly improved by shot-peening. After being formed and heat-treated, the leaf is cold-straightened as indicated. The results will be good or bad depending upon the direction of "cold-straightening" with respect to external working load. When the straightening load is taken off the leaf spring is released and residual stresses as indicated at (C) in Fig. 6 are developed. If the upper surface of the spring is now shot-peened, the compressive stresses developed thereby greatly increase the compressive stresses in the upper layers as indicated at (D). When the spring is then subjected to working loads, the stress distribution is as shown at (E). The conventionally calculated stresses are indicated by the dotted line. It will be observed that the calculated stresses are greater than the actual stresses and that the residual stresses are beneficial.

If the surfaces of any part subjected to bending be cold-worked by shot-peening or other methods, the surface layers are compressed. Subsequently applied working loads then produce stresses that algebraically add to the residual compressive stresses, so that the final maximum tension stress is correspondingly less than what it would be without the residual stress and the maximum compressive stress is correspondingly greater. However, fatigue failure is never caused by compressive stresses; the cause is always and only tensile stresses. Hence although the residual compressive stresses increase the final working compressive stresses, the fatigue life of the part is increased because the tension stresses have been reduced.

In tension members subjected to cyclic loading, if the material is sound the fatigue cracks always start at the surface. By re-

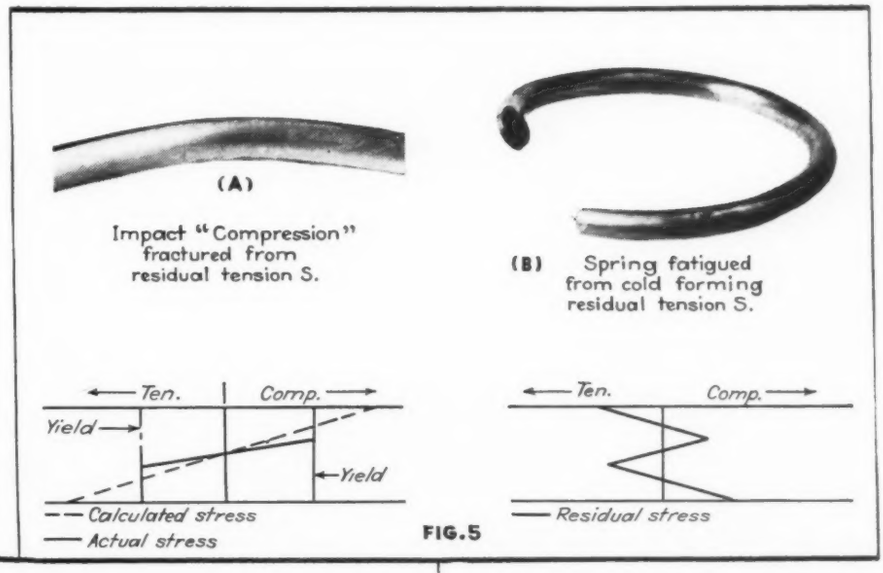


FIG. 5

Fig. 5—Helical coil springs that failed because the cold-forming load was in a direction different from the subsequent working load.

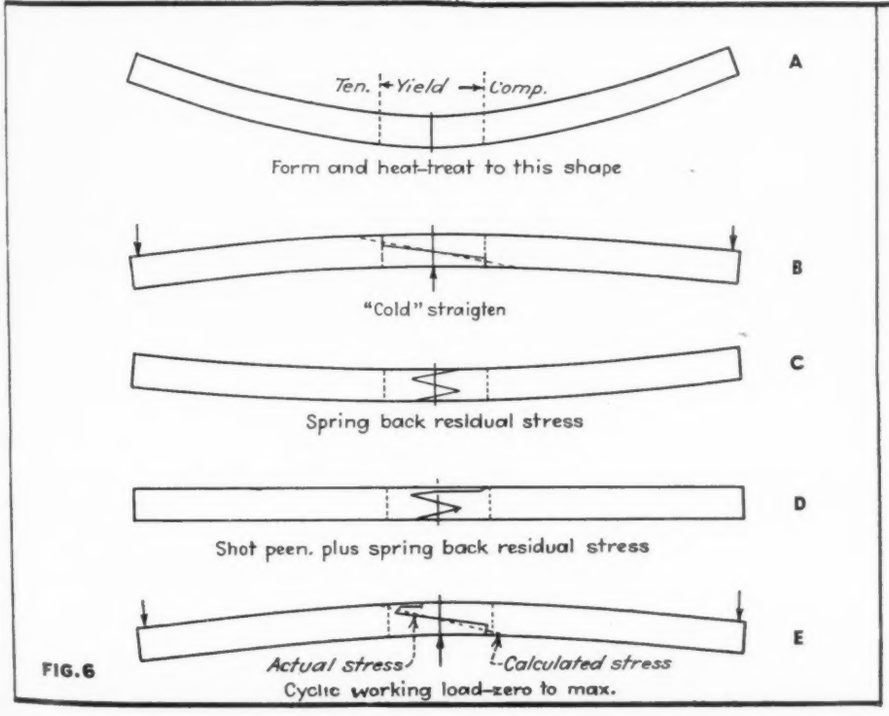


FIG. 6

Fig. 6—Residual and working stresses produced in a leaf spring in the manufacturing operations and in service. The dotted lines indicate the calculated stresses and the solid lines the actual stresses developed. At (B) the tensile and compressive stresses produced in cold straightening exceed the yield point. At (C) the leaf has sprung back after the release of the straightening loads thereby creating the residual stress pattern indicated by the solid line. At (D) shot-peening of the upper surface increased the compression stress as indicated. At (E) the application of the working load results in the stress pattern shown, the tensile stress in the upper shot-peened surface being only a fraction of the stress that would have been developed if the surface had not been shot-peened.

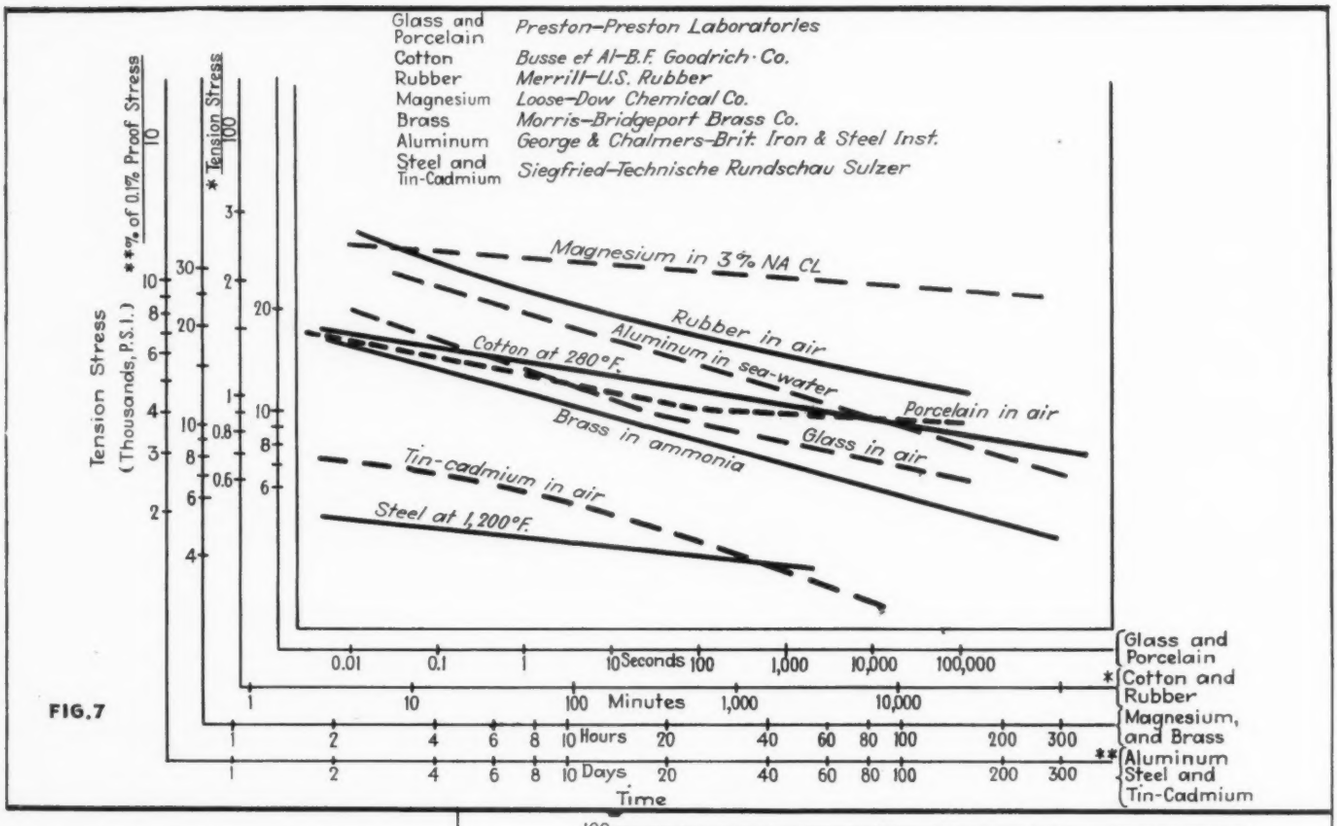

Glass and Porcelain — Preston–Preston Laboratories
Cotton — Busse et Al–B.F. Goodrich Co.
Rubber — Merrill–U.S. Rubber
Magnesium — Loose–Dow Chemical Co.
Brass — Morris–Bridgeport Brass Co.
Aluminum — George & Chalmers–Brit. Iron & Steel Inst.
Steel and Tin-Cadmium — Siegfried–Technische Rundschau Sulzer
Tension Stress (Thousands, P.S.I.)
**% of 0.1% Proof Stress
* Tension Stress
Magnesium in 3% NA CL
Rubber in air
Aluminum in sea-water
Cotton at 280°F.
Porcelain in air
Glass in air
Brass in ammonia
Tin-cadmium in air
Steel at 1,200°F.
0.01 0.1 1 10 Seconds 100 1,000 10,000 100,000
1 10 100 Minutes 1,000 10,000
1 2 4 6 8 10 Hours 20 40 60 80 100 200 300
1 2 4 6 8 10 Days 20 40 60 80 100 200 300
Time
Glass and Porcelain
* Cotton and Rubber
Magnesium, and Brass
** Aluminum
Steel and Tin-Cadmium

FIG. 7

Fig. 7—Tension stress corrosion of various materials in various environments. The horizontal scales give the time in which fracture occurred when the material was subjected to the corresponding stress indicated by the ordinates.

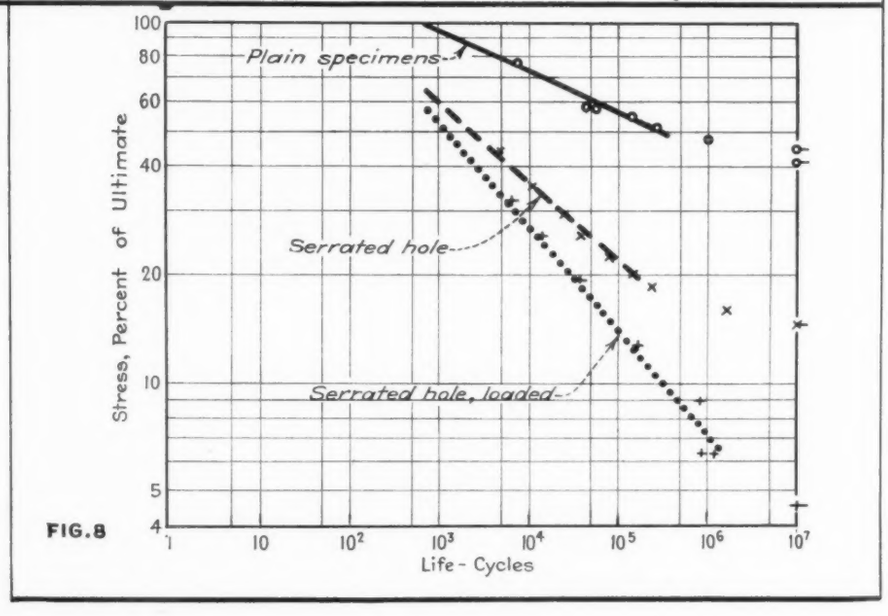

Plain specimens
Serrated hole
Serrated hole, loaded
Stress, Percent of Ultimate
100 80 60 40 20 10 5 4
1 10 10^2 10^3 10^4 10^5 10^6 10^7
Life - Cycles

FIG. 8

Fig. 8—Effects of stress raisers on the fatigue life plotted to log-log scales.

ducing the surface tension stresses the fatigue life of the part is increased. Incidentally, cold-working does not appreciably increase the static strength of the material insofar as its true physical properties are concerned.

In every design the residual stresses in the part will be dangerous or advantageous depending on their orientation with respect to the stresses developed by the externally applied loads. It is up to the designer to analyze each problem correctly and to take advantage of the increased fatigue life obtainable by residual stresses and to avoid or eliminate residual stresses if they are going to be harmful.

Failure by Stress Corrosion

In some design problems, although the stresses as calculated by the conventional formulas may be substantially correct and well within the allowable limits as indicated by the physical properties of the material, the part might fail by stress corrosion. This occurs when a surface under severe static tension is exposed to corrosive media, which may be anything from air to acids. The curves in Fig. 7 show the life of various materials with respect to failure by stress corrosion. The ordinates indicate the surface static stress in tension and the horizontal scales indicate the time in which ultimate failure occurs. It is highly significant that even glass in air suffers from stress corrosion. By peening or otherwise cold-working the surface the tensile stresses can be reduced or eliminated on the surface and thereby stress corrosion will be prevented.

In mechanical engineering design, possible failure by fatigue is much more to be guarded against than in civil engineering designs. The latter designs usually have such high "factors of safety" that the actual working stresses are but a relatively small fraction of the ultimate strength of the material, and complete stress reversals are rarely encountered. In machinery design the working stresses must be high, otherwise weights would be prohibitive. Therefore, machine designers must be extremely careful to avoid conditions that create stress raisers.

The effects of stress raiser plotted on a

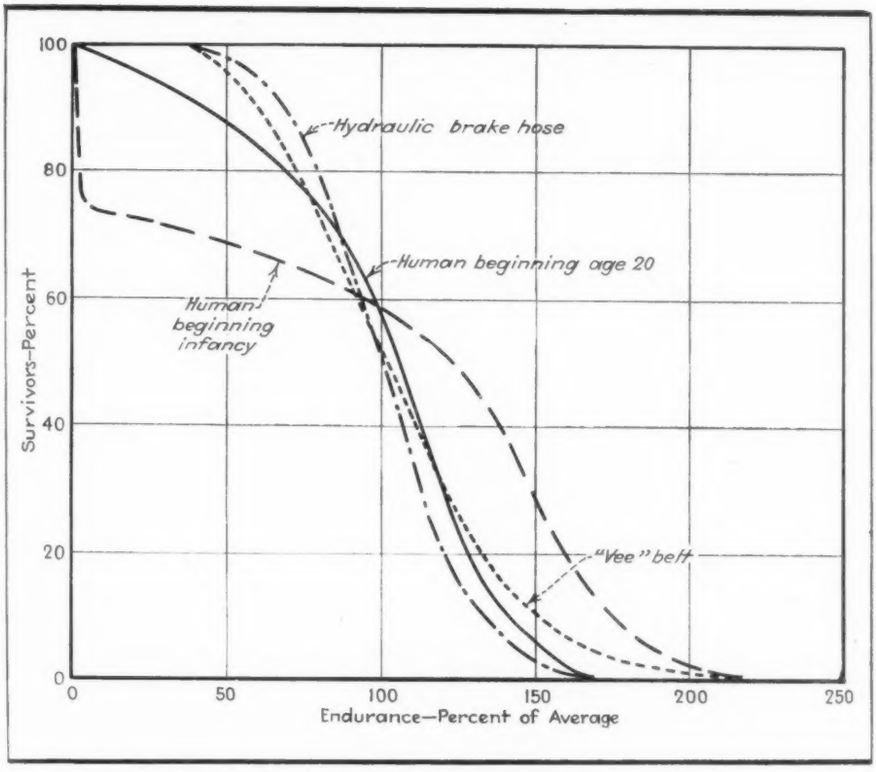


Fig. 9—When the endurance life of a large number of samples is plotted as a percent of the average life the resulting curve is similar to the human life expectancy curve beginning at age 20. The life expectancy curve for humans beginning at infancy drops sharply at its beginning because of high infant mortality.

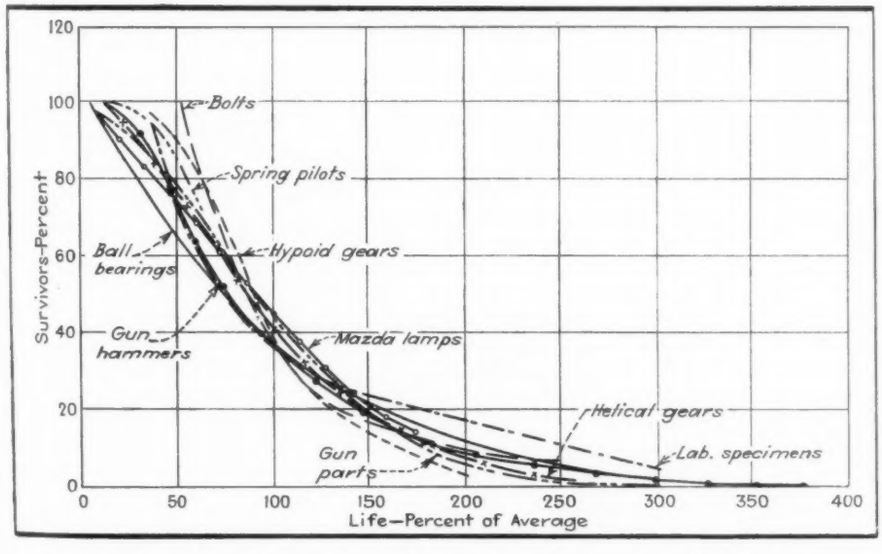


Fig. 10—Life curves of various manufactured products as based on the average life of that product. The curves are all similar to each other and to the life expectancy curve of humans and typical of the "probability and chance" curve.

log-log scale are shown in Fig. 8. The stress is plotted as a percent of the ultimate strength of the material. As indicated by the curves at 10,000,000 cycles the fatigue strength of the part with a serrated hole is only about one fifth of that of the plain specimen. This is indicative of the extent to which the machine designer will go astray if he ignores the effect of stress concentration.

Regardless of perfection of design and manufacture, any and every part will eventually fail. If the life of a great number of the same part be plotted the curve obtained will have the same general form as the curve for the life expectancy of humans, if the curve for the latter is started at the age of 20. This is shown by the curves in Fig. 9. The reason for the different trend of the human life expectancy curve taken over the whole of the span of life beginning at infancy, from that beginning at the age 20, is accounted for by the high infant mortality rate. Thus we have the principle that for every machine part, the life expectancy of the part follows a definite trend. This is also shown by the chart in Fig. 10. It is clearly indicative of the fact that the law of probability and chance applies in mechanical engineering design just as it does in all natural operations.

Limitations of Design Formulas

In the textbooks and teachings of mechanical engineering design, the major emphasis is usually placed on the type of failures produced on standard testing machines. Records of countless service failures clearly show, almost without exception, that the primary cause for the failure was one or a number of conditions or factors that are not duplicated in tests on standard specimens and that are not embodied in the design formulas. Thus the "contributing conditions and factors" become the determinants of the life of the part, and because they vary quantitatively and qualitatively from piece to piece, the life of the individual pieces follows the basic life expectancy curve.

Each of the separate elements that make up a machine will have a certain life expectancy. Thus it is that when a machine is produced in large numbers, such as automobiles, the order of succession of part failures as the machine grows older, differs for the various cars. In one car of a given make and model the valve spring might be the first part to fail. In another it might be an axle shaft or a connecting rod, or some other part. Usually each car has a happy distribution of long life and short life elements, but there may also be an unusual car that has all long-life parts. Such a car will probably wear out without breakage. And then there is the equally rare car that has a preponderance of short life pieces and "everything" starts breaking of failing early in its life. It is the "cluck" car, and by virtue of steadily improving design and production it is steadily becoming a greater rarity.

Only a few of the many known causes for failure or possible failure have been mentioned. Undoubtedly as time goes on additional knowledge concerning the behavior of materials, the effects of stresses and the causes contributing to failures will be discovered. But even as of today we have already progressed in mechanical engineering design far beyond the field of civil engineering structural design. It is a fact that should be properly recognized in our engineering schools and the textbooks and teachings should be patterned accordingly.

Design of Weldments For Best Results at Lowest Costs

DR. I. A. OEHLER
Director of Research, The American Welding & Manufacturing Company

How the use of standard and special rolled shapes, proper shop equipment and sound design techniques achieve superior parts at lower costs. Salient features of good welding design are set forth and their application in the design and fabrication of a few parts are discussed.

WELDING should be looked upon not as a substitute for some other methods of joining but rather as a means for achieving a structural design that will meet its functional requirements most advantageously and at a minimum cost. This approach to the problem has frequently resulted in the creation of new constructions and redesigns of existing structures with appreciable savings in weight, lowered costs, or higher functional efficiency.

To develop the best design of a structure it is necessary that the design engineer be thoroughly familiar with all the possible methods of fabrication and joining and that he have a comprehensive knowledge of the advantages and limitations of each type. With reference to the welding method to be employed, the designer must be equally familiar with all the different types including electric arc, gas welding and resistance welding. He must also be famliar with the various factors that contribute to the manufacturing costs in order that he be able to select not only the most economical method but also the most economical use of the method selected. This last requires a knowledge of the relations between a weld location and weld cost.

Because standard and special rolled shapes were originally designed to give a most advantageous distribution of material over the cross section for specific kinds of loads, they are commonly used in designs of weldments. The highly efficient material distribution in standard shapes makes it possible for the designer to develop a structure that will require a minimum of fabricating operations and minimum of welding with corresponding low costs.

However, it should not be assumed that standard shapes and plates should be used wherever possible. Often a complete analysis of the design will show that a forging or a casting will be more economical than the use of standard shapes and plates. Also, it will sometimes be found that simple bending operations on flat plates will produce a member having the desired strength and stiffness thereby avoiding the cost of welding that would result from building the cross-section with plates and shapes. In every design the engineer should endeavor to keep the amount of welding at a minimum through the use of bending and other simple forming operations, these operations costing relatively little.

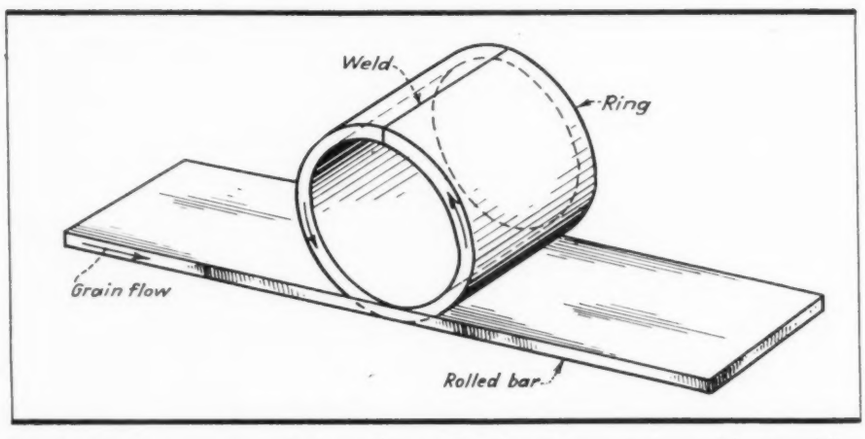


Fig. 1—Bearing races made by rolling a flat bar into a circle and butt welding the ends have the grain flow in the advantageous circumferential direction.

When the items are to be produced in relatively large quantities it will often be found that some of the electric resistance welding methods will be most advantageous. Spot welding is extremely fast and in many applications furnishes all the strength required of the joint. Recent development in the art now enables the use of spot welding for almost any material and has greatly increased the maximum thickness of sheets that can be spot welded economically. By embossing projections in the proper positions parts can be spot welded without jigs with the assurance that all the joints will be exactly alike with reference to the relative location of the pieces joined.

Another significant development during the war period has been the extension in the field of application of resistance butt welding. Cross-sections up to and including 6 in. in diameter are now resistance butt welded in commercial production.

Wheel rims and other types of rims have been made for many years by simply rolling a bar into a circle and resistance welding the ends together to form a continuous circle. Rings of heavy cross-section can be produced in this manner by the use of large capacity welding equipment now available.

Perhaps the most significant development affecting the field of resistance welding has been that of the full automatic welders. These machines can be preset to give a selected electrode pressure, welding time, and welding energy. Thus, all welds are made as nearly exactly the same as possible.

It is impossible in a brief article to set forth all the factors that must be considered when the engineer develops his welded structure. However, to bring out some of the salient features of good welding design, the design and fabrication of a few parts will be described and discussed.

Bearing races were formerly made either by forging or by machining the race from

seamless tubing. The selection of the method used depended upon the diameter and cross-section of the race.

Obviously the direction of the grain in a ball race will be an important factor in the load carrying capacity and durability of the unit. When made from a forging or from a piece of seamless tubing the grain direction was either parallel or radial to the central axis of the ring. It is obvious that if the grain direction were circumferential, it would be an appreciable improvement.

To achieve the desired circumferential direction of metal grain flow, the bearing race is now produced by rolling a piece of flat bar stock into ring form and flash butt welding the ends together to form a continuous circle, thereby forming the race as illustrated in Fig. 1. After butt welding, the ring is sized and trued to the desired diameter on a fixture. It is then machined, heat-treated, and ground. It has been found by tests and performance records that bearing races made by this method possess superior properties and longer life than those made in the conventional way, the improvement being attributed to the circumferential direction of metal grain flow.

Frames for Generators and Motors

Quite a few years ago, manufacturers began to make frames for d.c. generators out of flat plate rolled into a cylindrical form and the joint welded by one of several means. Here again there was a distinct advantage in having circumferential grain flow, in this instance because of improved magnetic qualities. However, there was another and perhaps more important advantage in the formed-plate welded generator frames. This method of fabrication permitted the use of special ingot iron plate that possesses superior magnetic properties, which could not be duplicated in forgings, castings, or seamless tubing. Superior magnetic properties of the special ingot iron plate permit a reduction in the cross-section of the motor frame as compared to the minimum cross-section that would be permissible if the frame were made of forged or cast steel or seamless tubing.

Aircraft Engine Crankcase

Conventionally, aircraft engine crankcases were produced entirely as forgings made of chrome-moly steel and heat-treated to increase the physical properties of the material. Because of the unusual shape of the finished section, much of the forgings had to be removed by machining. This was a long and costly process. Naturally much thought was given to the possibility of designing the crankcase as a weldment.

In Fig. 2 is shown the final design for a selected aircraft engine crankcase. It is built up from two components welded together as indicated. The ring section is produced by bending the rolled special channel section into a circle, the joint flash butt welded and the ring then formed to an exact circle of correct diameter and contour by shrinking it on a mandrel on a segmented hydraulic shrinker. The diaphragm is a forging. The edges of the ring and diaphragm were machined in preparation for welding and the two parts were joined by the submerged arc process, using an alloy welding rod. It was necessary to preheat the ring and keep the diaphragm cool during the welding operation so that when the weld began to contract upon cooling, the ring would shrink with it, thus keeping the weld under compression and avoiding rupture in the weld.

Openings in the ring for the cylinders were torch cut to almost the final dimension. The part was then heat-treated and finish machined.

Whereas the former design of aircraft engine crankcase was an expensive heavy forging that required a great amount of machining, the welded design is a relatively simple forging and a formed mill section, the two joined by controlled welding and requiring a minimum amount of machining. Expensive and extensive tests have proved that in every respect the weldment is equal to the forged crankcase in its functional performance.

Truck Side Frames

Minimum weight is greatly desired in side frames on all types of railroad cars. The conventional side frame is a one-piece steel casting. In service the frame is loaded in bending with the load applied over about the middle third and transmitted through the frame to the pedestals near the ends. Thus the top chord is in compression and the lower chord in tension. Between the chords is a web section.

In the welded design as finally developed, the top chord is a tubular compression member made of a standard tube, which

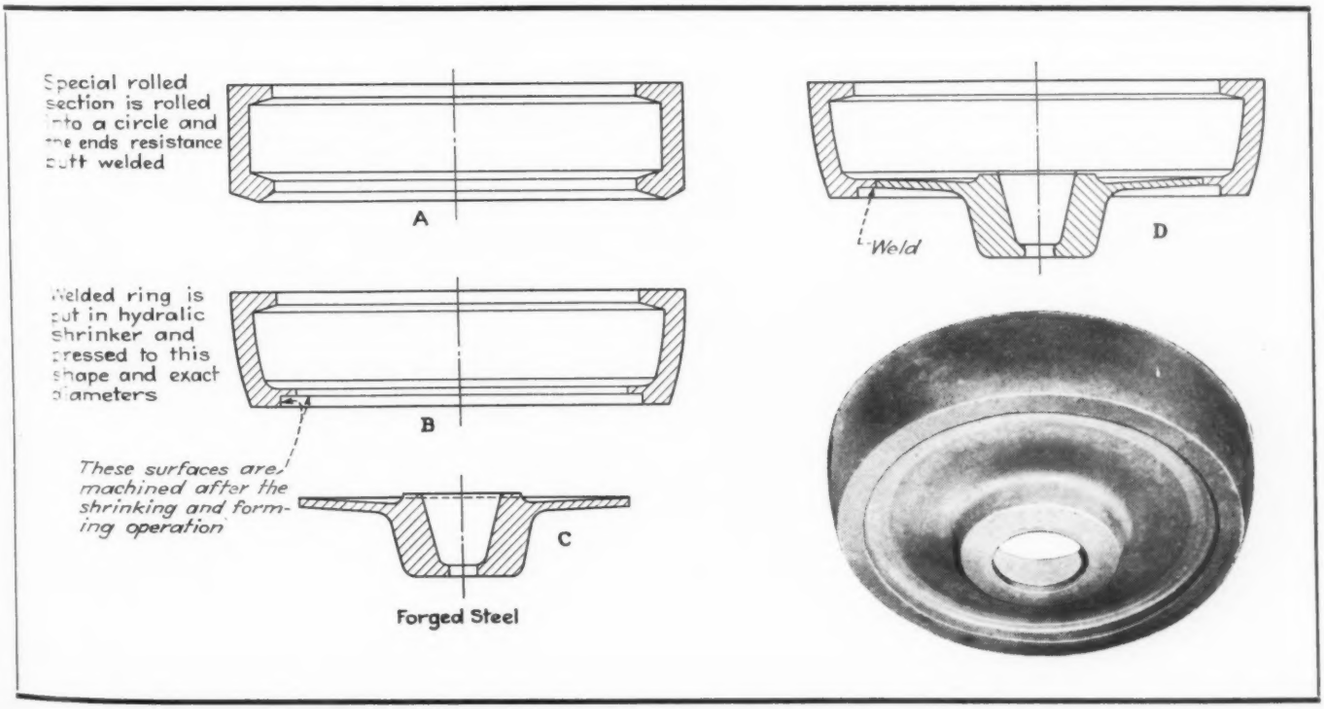

Special rolled section is rolled into a circle and the ends resistance butt welded
A
Welded ring is put in hydralic shrinker and pressed to this shape and exact diameters
B
These surfaces are machined after the shrinking and forming operation
C
Forged Steel
Weld
D

Fig. 2—Welded engine crankcase. The finished weldment consists of the rolled channel section and the forged diaphragm.

requires only a slight amount of forming. The web section is steel plate that is torch cut to shape. The bottom chord is a rectangular bar serving as a tension member. The upper tubular member and the rectangular bar are both secured to the plate that serves as the web. In the area of high shear, namely near the ends of the side frame, plate pads are welded to the main plate web to serve as reinforcement.

This welded side frame represents a combination of ideally suited standard sections for carrying the various loads imposed. The result is a weldment fully capable of performing its function and effecting a great saving in weight.

In the design of any beam section the designer is frequently tempted to use a deep rolled channel with a cover plate welded to the edges of the flanges as shown in Fig. 4. Such a design, however, puts the weld in an area of high stress, the weld metal being close to the region of maximum compressive stress if it is at the top of the beam or in the region of maximum tensile stress if it is at the bottom of the beam. A greatly superior construction is obtained by using two channels and welding the edges of the flanges together as also shown in Fig. 4. In this latter construction all of the weld metal is in the neutral plane and not subjected to high stresses. The only stress to which the weld metal will be subjected would be a shear load perpendicular to the line of the weld and the horizontal shear, which loads are usually not significant.

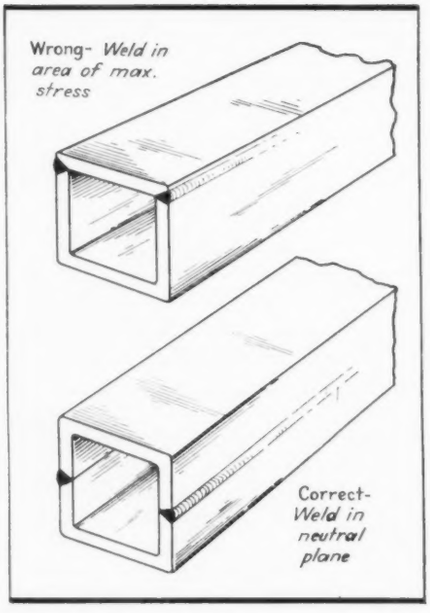


Fig. 4—Correct and incorrect weld positions in built-up beams.

A few precautions should be taken whenever a box section is built up as shown in Fig. 4. If the weld metal does not penetrate all the way through the joint, it is likely to result in reentrant angles that become points of stress concentration. This will not be important in the double channel construction shown in Fig. 4, if the member is used as a beam. If, however, the member is also subjected to torsional loads it is possible that a fatigue crack will develop.

It is also interesting to observe that whereas the double channel construction is better for beams, the construction using the single deep channel and cover plate is the better if the member is to be subjected to torsion. This is because the shear stresses in the rectangular member subjected to torsion are zero at the corners and a maximum at the midpoints of the sides.

Reducing Die Costs by Weldments

Frequently it develops that appreciable production savings can be made in assembling a number of separate units by resistance welding. It is for this reason that a design should be carefully studied to determine whether or not all of the sections of the element must of necessity be made by the same process. Another factor to consider is whether or not the manufacture of a member in two or more separate parts will reduce die costs. In Fig. 6 is shown a simple track tool that is nothing more than a rod

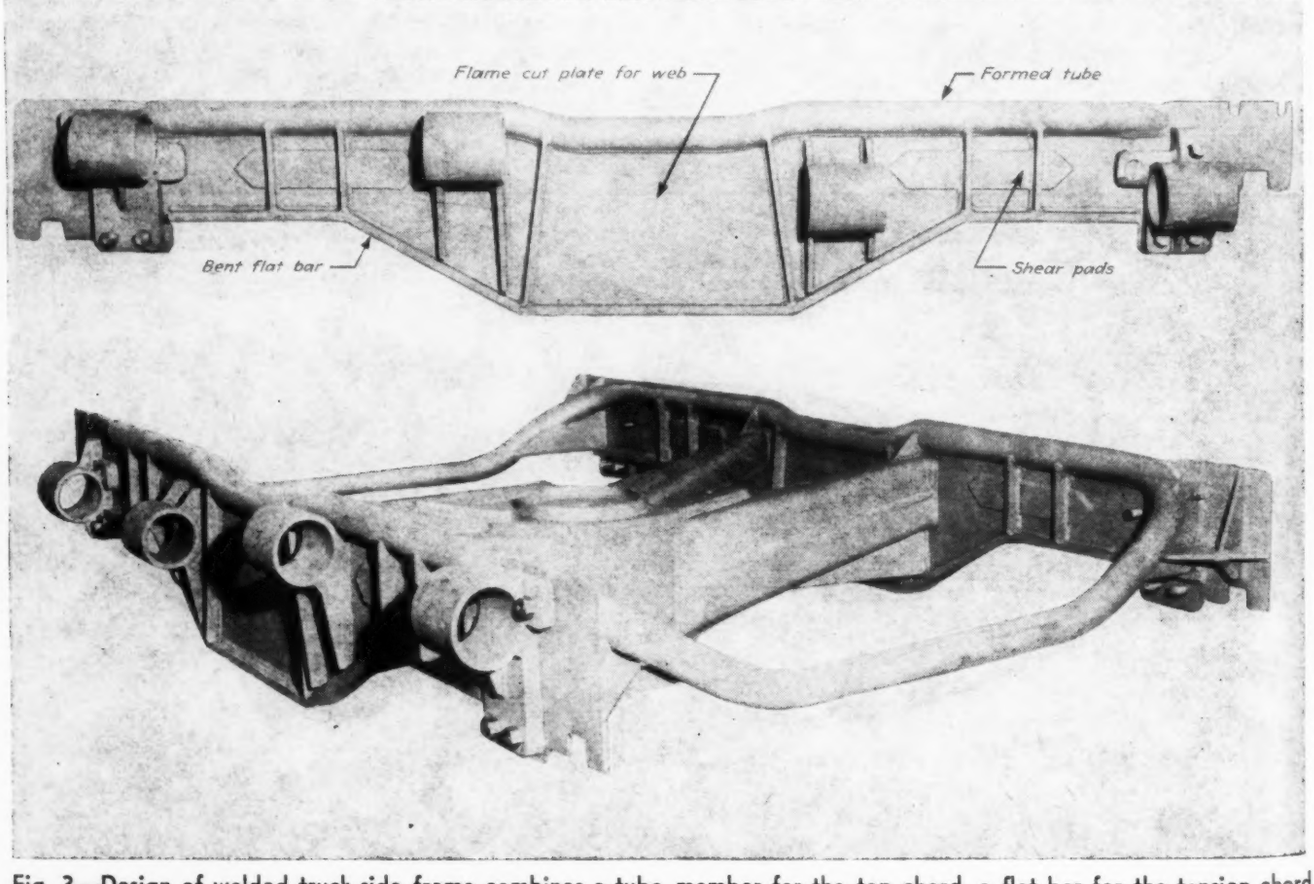


Fig. 3—Design of welded truck side frame combines a tube member for the top chord, a flat bar for the tension chord at the bottom, with a flame cut plate serving as the web.

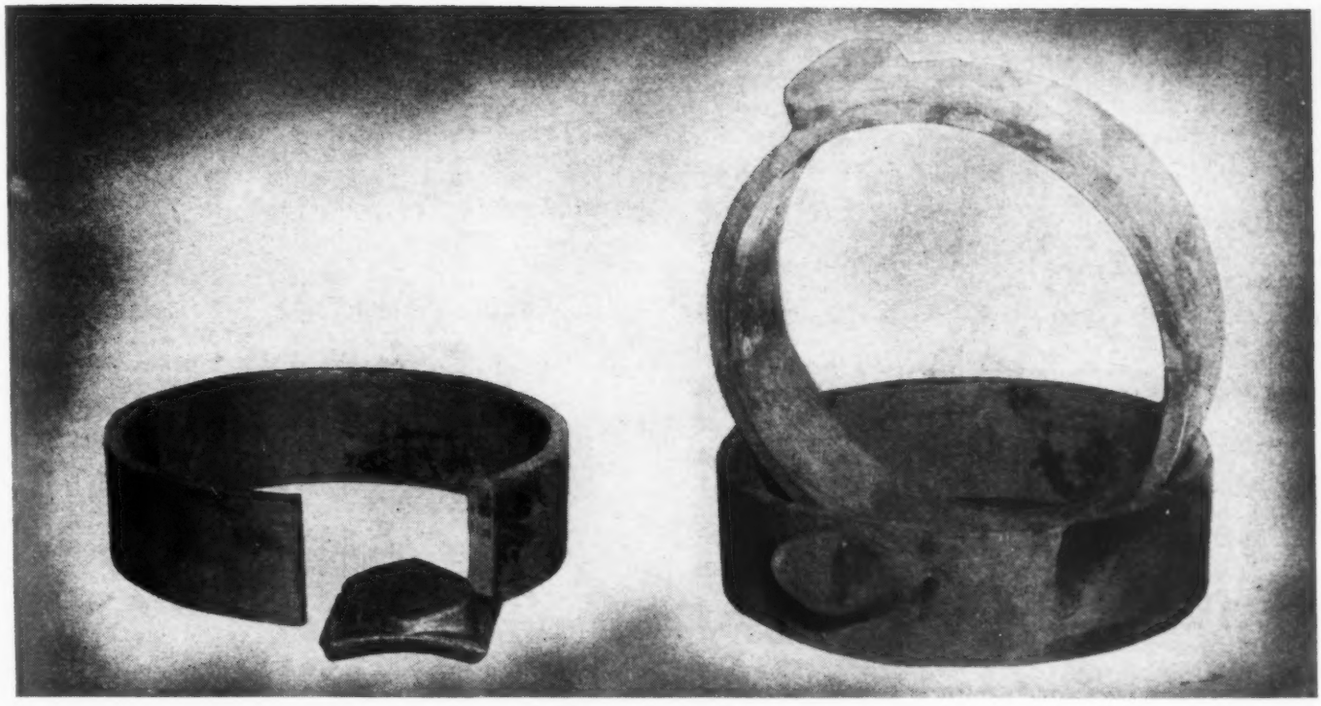


Fig. 5—Diesel engine water inlet ring fabricated from a simple drop forging and a formed steel bar. The bar was bent into circular form on a three roll machine, with gap for insertion of forging. Original welded design provided for arc welding, but flash butt welding the two joints on an automatic resulted in lower cost and better performance.

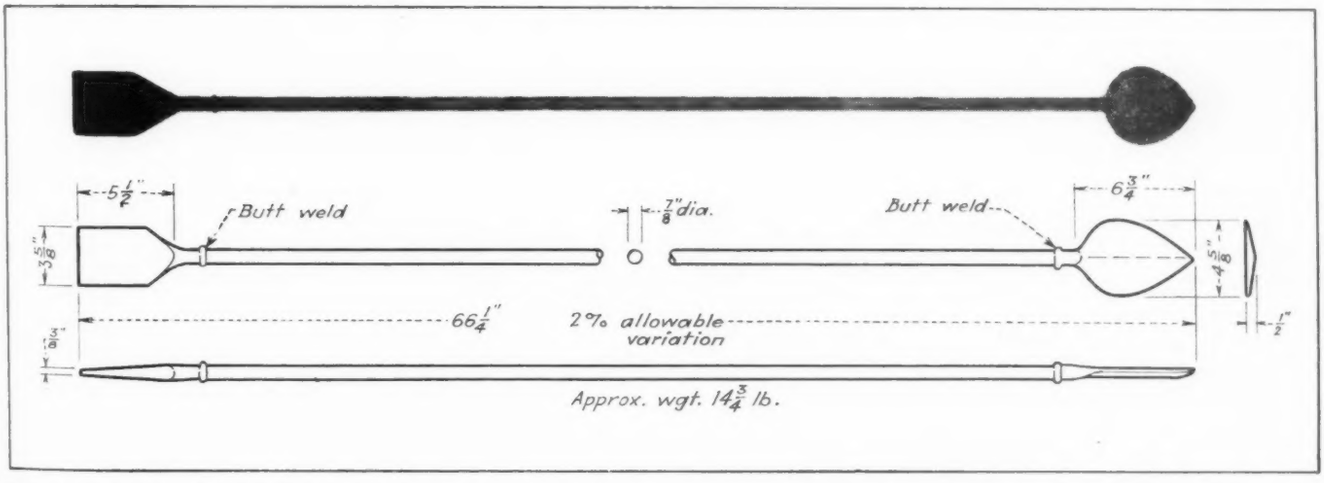


Fig. 6—To make this simple track tool as a one-piece forging requires many forging operations to upset the bar. Making it in three pieces and welding them together as shown, reduces cost considerably.

with a spade end forged at one end and a tamper at the other end.

Obviously there is no sound reason why the whole unit should be forged as a single piece at considerable cost. It is logically suggested that each of the ends be made as separate forgings. This was done and the spade and tamper ends were then flash butt welded to a round bar. This example shows that welding can be used to advantage together with other forms of construction and should not be looked upon as merely a competitive method.

Incidentally, if it had been desirable to reduce the weight of this tamper, a tube could have been used in place of the solid rod, a construction that would not have been feasible if the unit were made as a one-piece forging. However, in the particular tool illustrated, weight saving was not a consideration because the weight of the tool contributes to its proper functioning and hence a solid rod was used instead of a bar.

The foregoing examples serve to illustrate to a slight degree the wide latitude that a design engineer has when he is developing a design to be produced by welding. It is emphasized that welding should not be employed to the exclusion of more economical ways of arriving at a functionally satisfactory structure. It is generally cheaper to purchase standard rolled shapes than to have the shop make them as built-up welded structures. Sometimes it is cheaper to buy small or simple forgings or castings. And it is often better to bend or form shapes and plates rather than to butt-weld sections in order to accomplish the same result. By resorting to forming operations the welds can frequently be located in areas of low stress thereby saving weld metal and the necessity for close inspection. The deposited weld metal is more costly per pound than the same metal in any other form and therefore weld metal should be used sparingly.

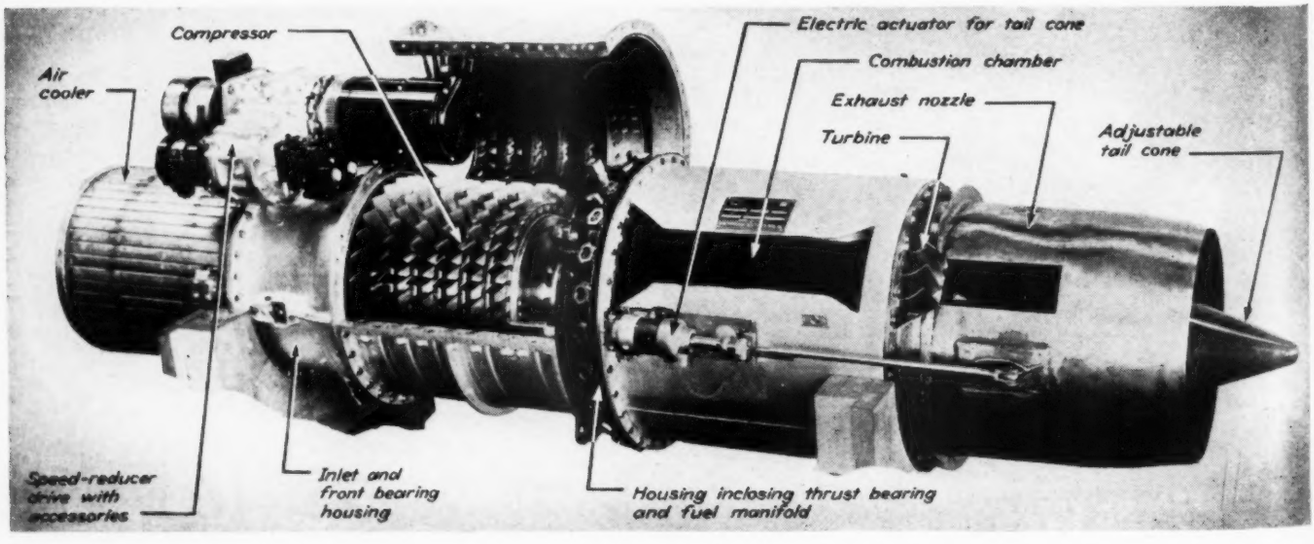
The illustrations, not described in the text, are self explanatory and serve to further emphasize the design possibilities through the use of welding when that process is used not to achieve a substitute but to create a better and less costly article.

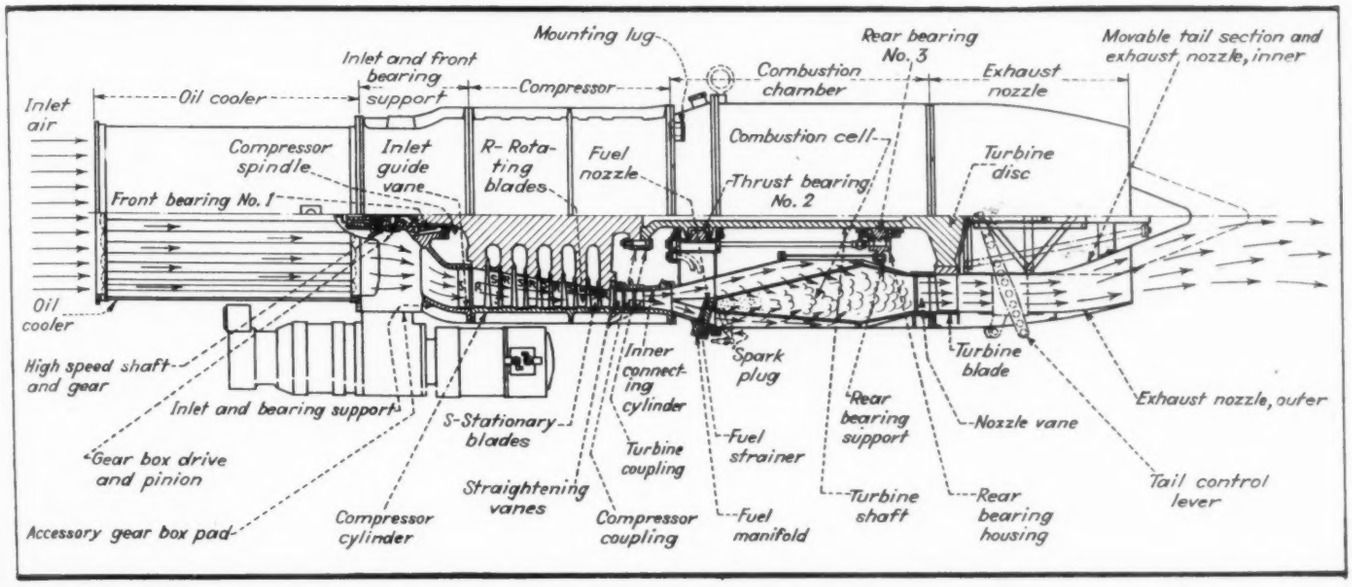
PRODUCT DESIGNS

Axial Flow Compressor

Reduces Jet Engine Diameter

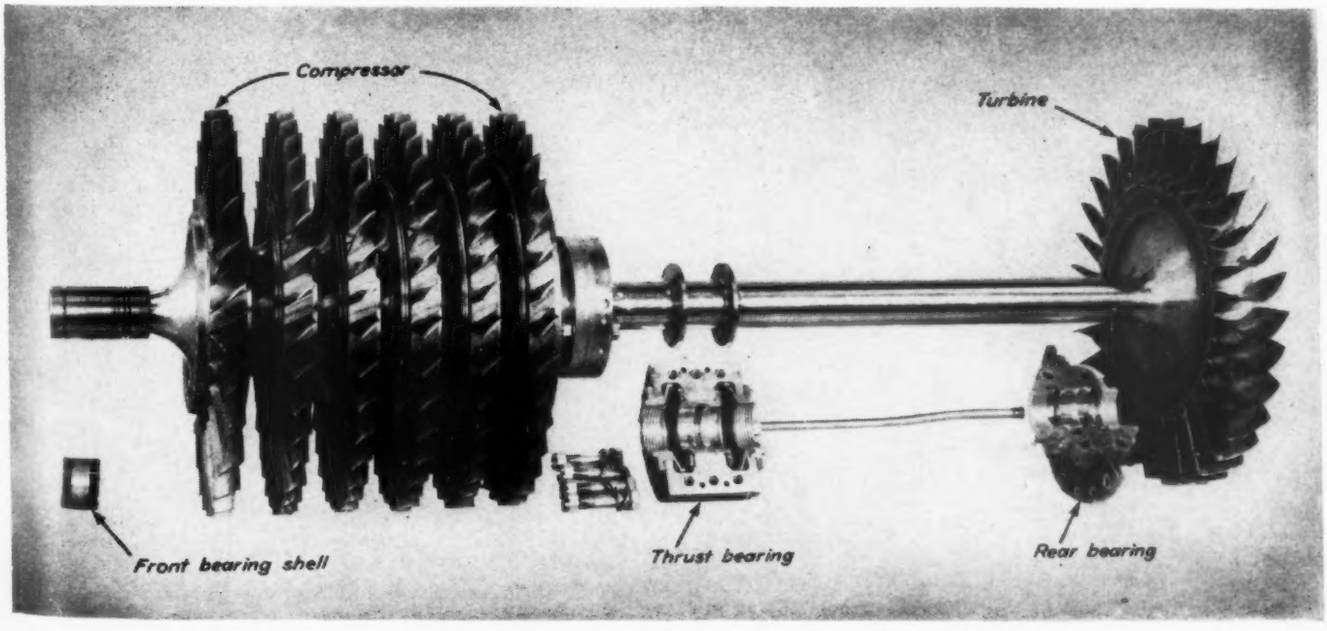
MINIMUM DIAMETER and weight with maximum thrust is the basic thought behind the design of the Yankee gas turbine for jet propulsion of aircraft. Using an axial flow compressor rather than one of the centrifugal type, this Westinghouse Electric Corporation engine of 21-in. diameter, less accessories, develops 1,400 lb. static thrust and weighs 826 lb. dry. About two-thirds of the total power developed is used to run the compressor. Net useful power at 375 m.p.h. is 1,400 hp. The engine can be used as the main power plant or as an auxiliary to a conventional engine. It consists of four basic units: Axial flow compressor, combustion chamber, gas turbine, and adjustable exhaust nozzle. The oil cooler, made of rectangular aluminum tubing, is forward of the intake. To facilitate clean design with small frontal areas, one accessory drive shaft extends from the main shaft to the speed reducer gear box located on the outside of the front bearing support. Mounted on the box are the lubricating and scavenge oil pump, tachometer generator, speed governor, fuel pump, starter control, generator and starter. Three bearings support the 6-stage compressor and single-stage turbine shafts. The installation is flexible because the engine can be mounted with the gear box up, down, or at either side.

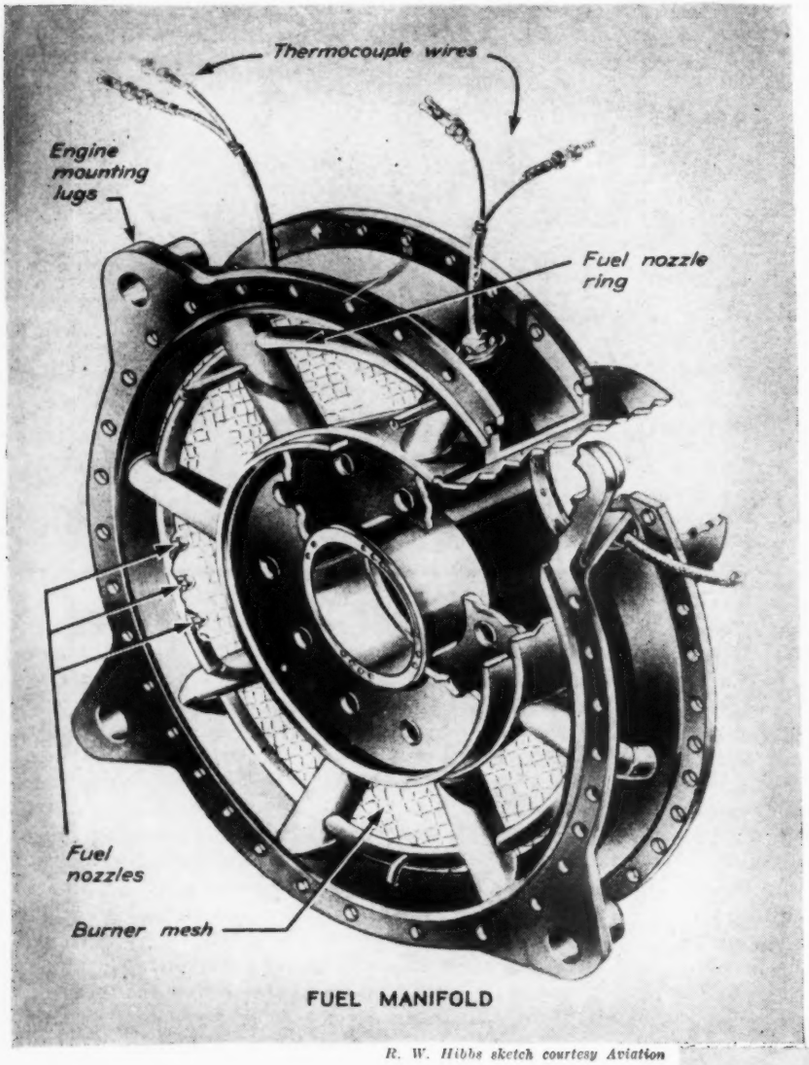




Inlet air rushes through oil cooler to enter inlet housed in front bearing support. Guide vanes at exit end of casting turn air to the correct angle for entering the first stage compressor blades. Pressure rise across each set of rotor-stator blades is approximately equal and total pressure rise is slightly over 3:1. Three rows of straightening vanes lead the air from the compressor to the combustion chamber. As air flows into the combustion chamber at the thrust bearing housing it passes through a screen of 6x6 mesh per sq. in. and around the fuel manifold made up of 24 nozzles into a perforated burner basket. This "burner basket" provides turbulence necessary for proper combustion at the high air velocities encountered. Two concentric conical sleeves running axially through the chamber fan out to form the "combustion cell." Fuel nozzles feed into the narrow end of the cell while perforations permit the air to enter along the cell walls. Thus combustion is virtually confined to the cell, while an envelope of cool air from the straightening vanes surrounds the combustion area and assists in cooling the combustion chamber walls. Hot gases from the combustion chamber pass through the nozzle assembly of 45 Vitallium vanes to turn the turbine wheel. Gases leave the turbine blading and pass into the atmosphere through the exhaust nozzle. This nozzle has an electrically controlled, inner adjustable cone that varies the amount of energy fed into the turbine with respect to energy left in propulsion jet. This is done by altering the exhaust area thus changing the turbine back pressure relative to atmospheric pressure. For maximum thrust the tail cone is moved aft. Air moves from the inlet through exhaust in 0.02 second.

Compressor spindle (below) including shaft at inlet end and coupling at discharge end is machined from single aluminum alloy forging, each disk having a double convex cross-section. Turbine shaft and disk are made from a one-piece forging, 19 percent nickel, 9 percent chromium stainless steel, with some tungsten and molybdenum added for hardness.

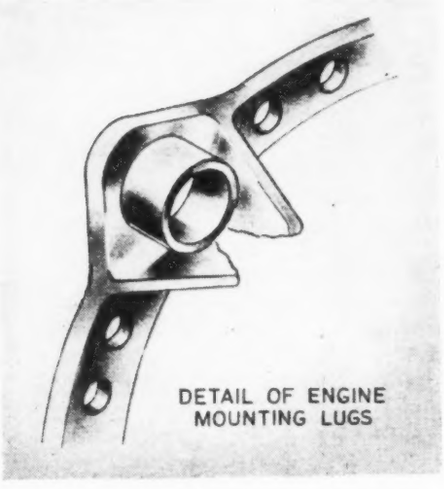




FUEL MANIFOLD

R. W. Hibbs sketch courtesy Aviation

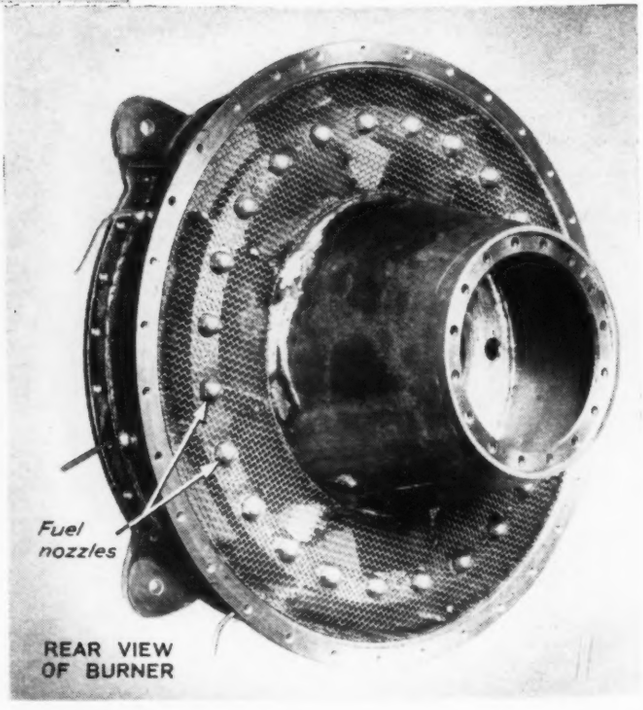
Fuel burner and thrust-bearing support are built up of three concentric stainless steel rings tied together with eight hollow struts. Outer ring has welded flanges at each end for bolting to compressor and combustion chamber; and flange at compressor end carries the four engine mounting lugs. Middle ring houses the 24 burner nozzles and supports the burner basket. Thrust bearing support is welded to inner ring. Burner basket mesh causes turbulence thus mixing air flow and fuel for combustion chamber. Seven of the eight faired hollow struts have passages for oil lines and thermocouples.



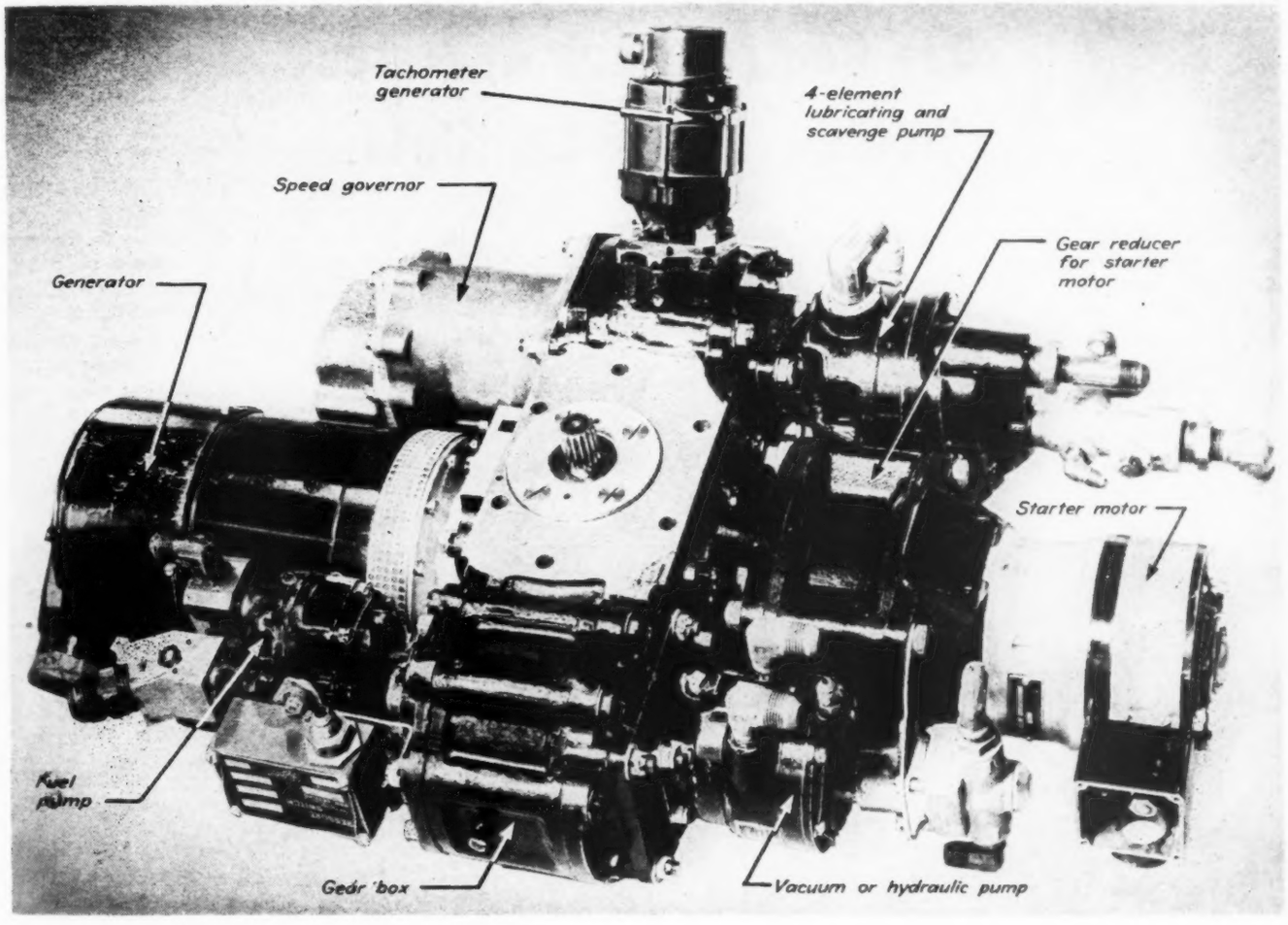
DETAIL OF ENGINE MOUNTING LUGS

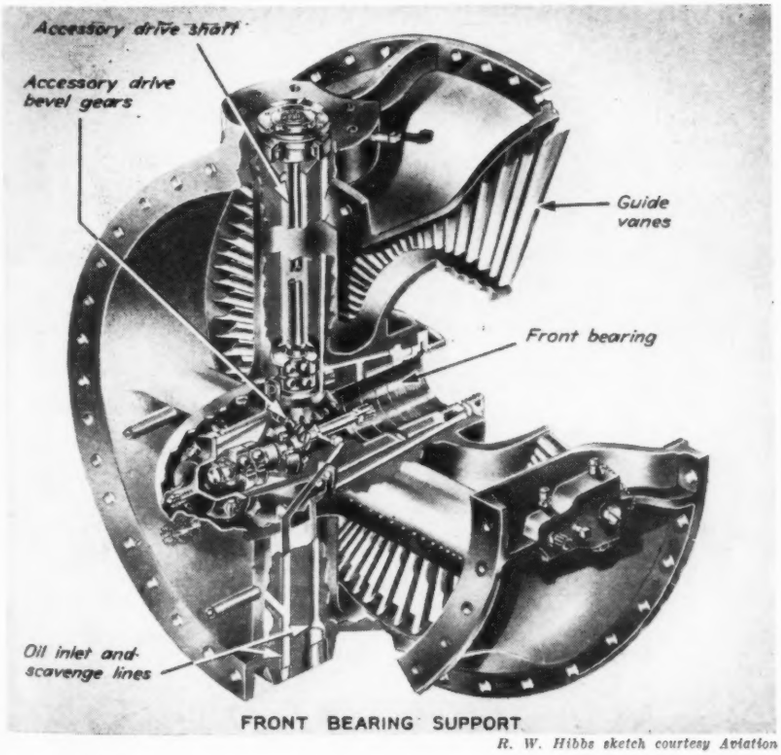
DETAIL OF COMPRESSOR AND TURBINE BLADING ASSEMBLY

Steel alloy blades with bulb-type roots are held in milled slots in the spindle disks by wire locking keys that are turned up, peened into the sides of the blade roots, and filed flush. Centrifugal stresses in the first compressor stage reach 50,000 G's. Normal temperature of gases entering the turbine nozzle vanes is about 1,200 deg. F.



REAR VIEW OF BURNER





FRONT BEARING SUPPORT

R. W. Hibbs sketch courtesy Aviation

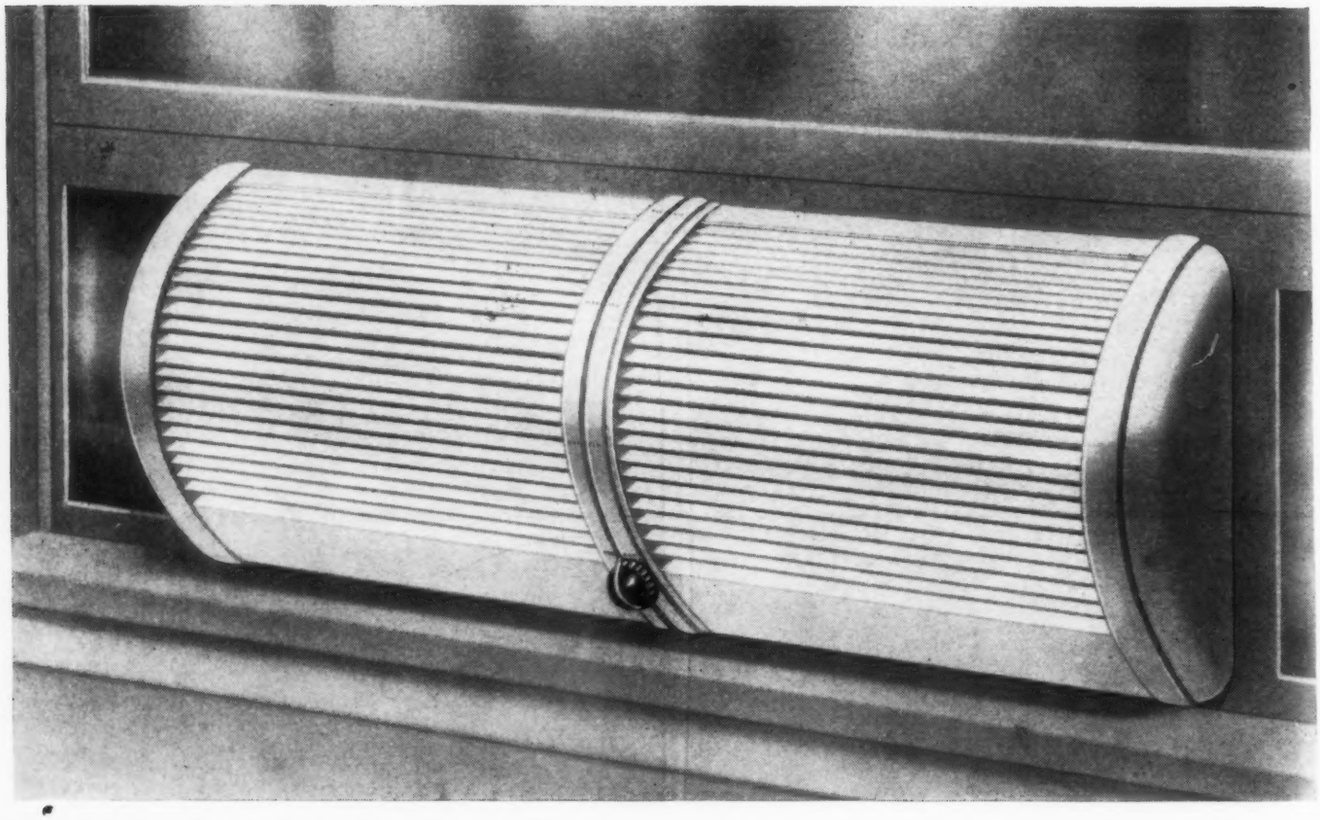
Engine accessories are driven from a single gear box mounted on the front bearing support. An aluminum casting houses the speed-reducer train of 12 gears. Electric starting motor has roller-type, free-wheel clutch and starts engine by driving main shaft through the speed reducer. Fuel pump is of positive displacement gear type to maintain fuel pressure at 6 to 15 lb. per sq. in. above atmospheric pressure at all times. Flyball-type governor actuates a valve that restricts fuel flow at overspeed conditions. Compactness and simplicity of accessory mounting and drive contribute to the clean design of engine.

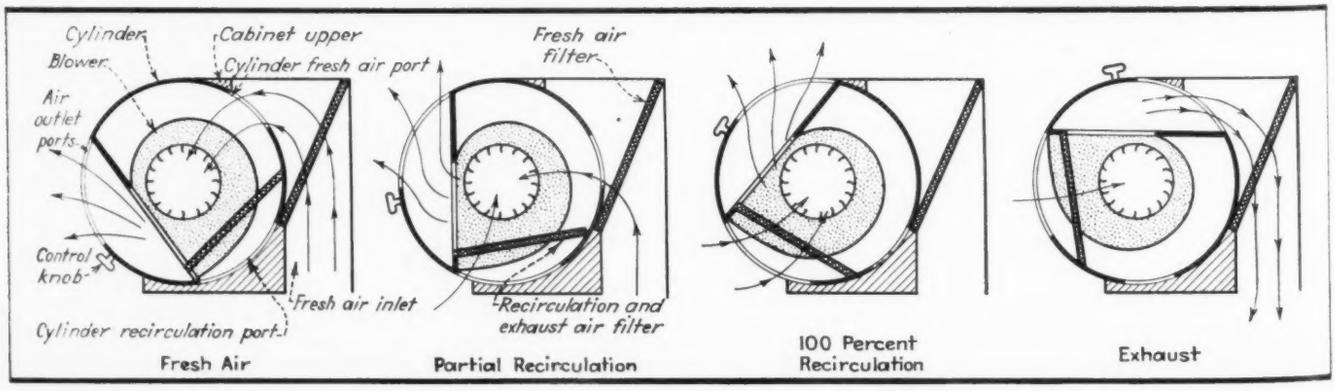
Front bearing support is an aluminum casting connected to the bearing housing by four struts. One strut houses the accessory drive shaft, others inclose the inlet and scavenge oil passages for main and bevel gear bearings. A spinner-type cap, detachable by removing one screw, gives access to accessory take-off drive bevel gears without further disassembly. Guide vanes at exit end of casting turn air to correct angle for entrance to first stage compressor blades. Thrust bearing, located between compressor and combustion chamber carries thrust and radial loads. Front and rear bearings are of the sleeve type and carry radial loads only.

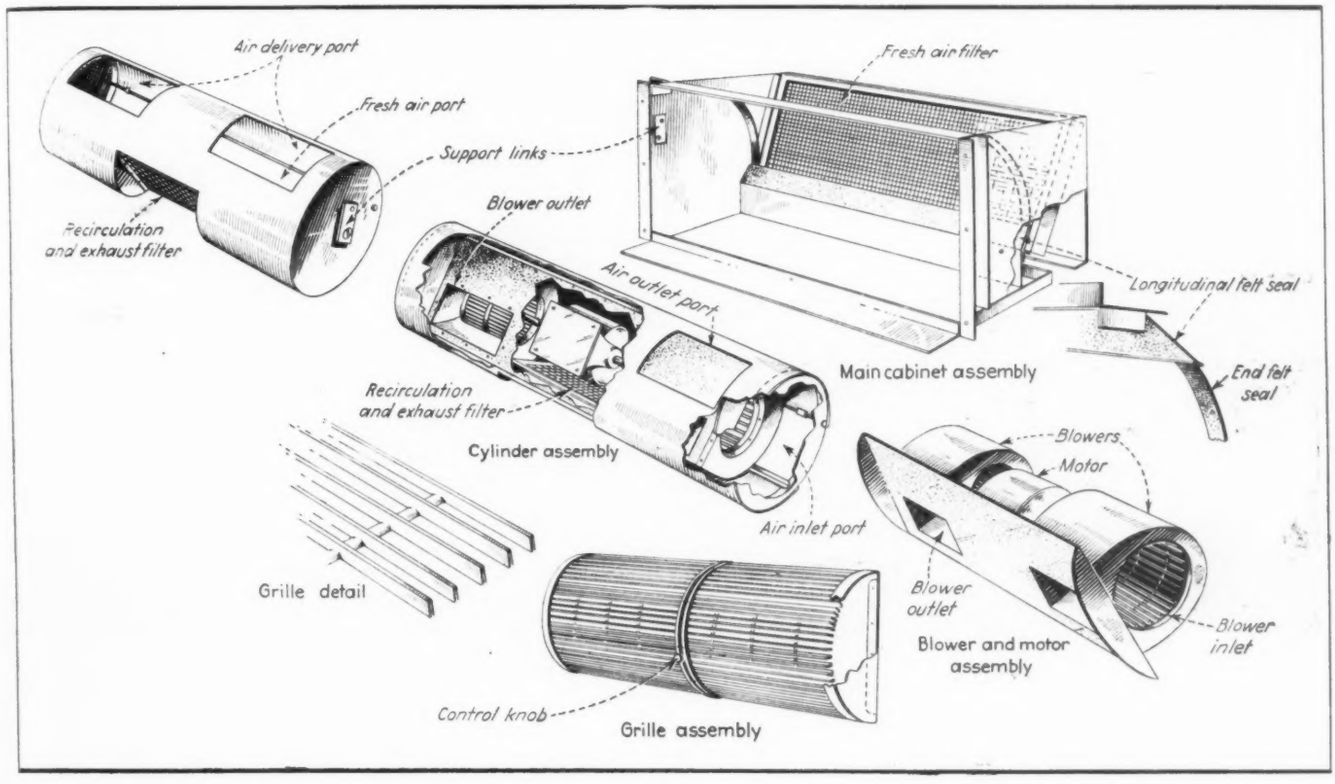
One Control Knob Operates Window Power Ventilator

SINGLE CONTROL KNOB operates all functions of a window power ventilator developed by Industrial Designer, Charles Davies, for United States Air Conditioning Corporation. Ventilator can bring in fresh air, recirculate part of the room air and mix it with fresh air, recirculate 100 percent room air or exhaust room air. Construction is of steel. Two filters; one for the fresh air and another for the recirculated air are used in the ventilator. Air volume is controlled by a rheostat that is part of the control knob. Design requires no dampers and is air-tight when not in use.

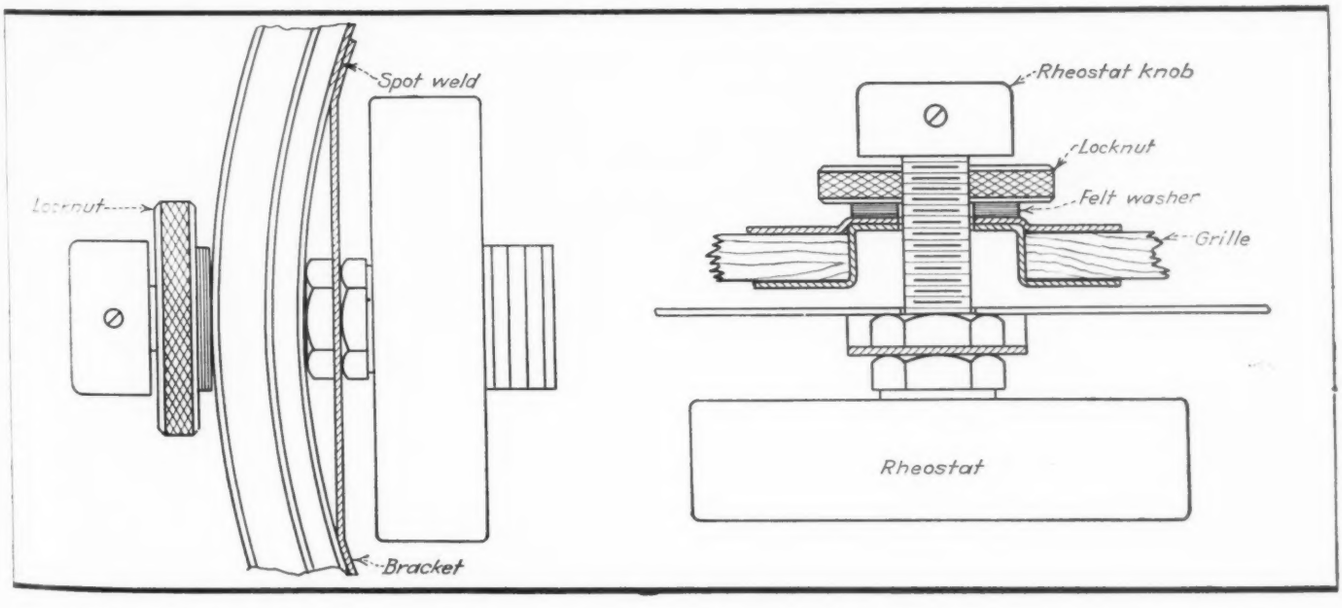
Four positions of the control knob select the different phases of ventilator operation. Bottom position draws fresh air through a filter to the blowers then out through the cylinder air outlet ports to the room. Moving the control knob to the first notch on the grille rotates the cylinder and partially opens the cylinder recirculation port thus, both fresh and room air are circulated through the ventilator. At the second notch the cylinder rotates sufficiently to close the cylinder fresh air port and thus, only room air circulates through the ventilator. Top position rotates the cylinder its maximum amount closing the fresh air inlet port. Room air is drawn into the ventilator through the cylinder recirculation port and discharged to the outside.



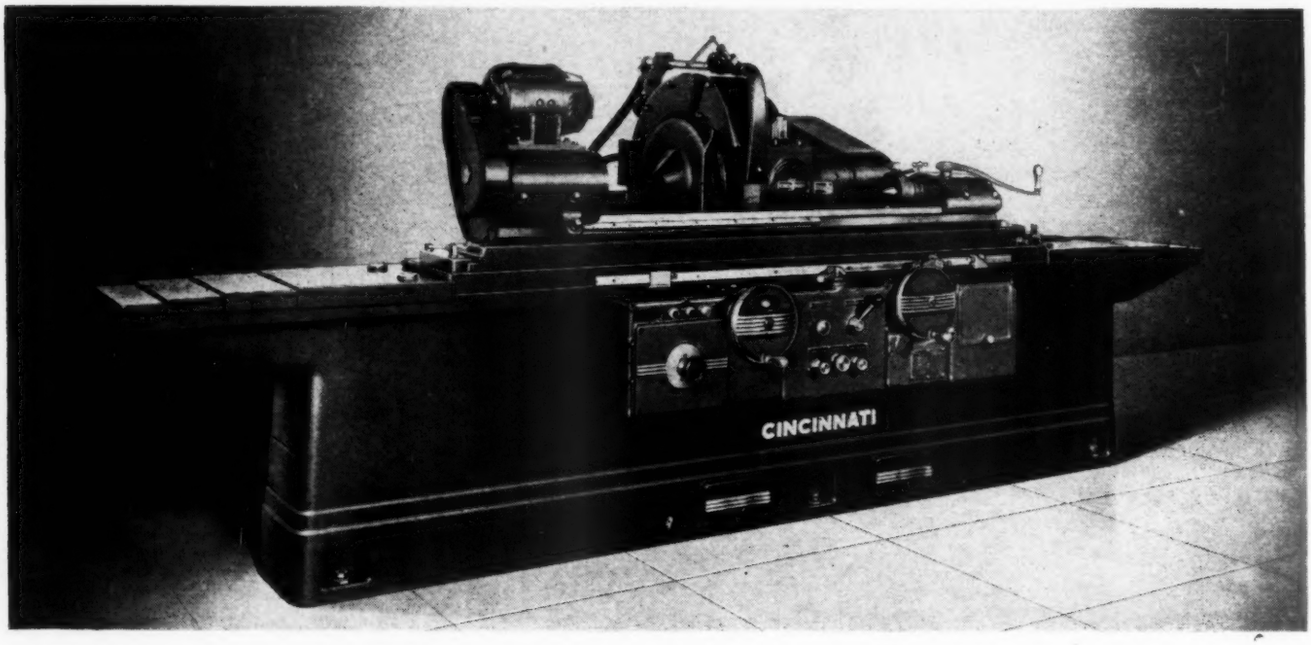




Ventilator consists of three major assemblies; the main cabinet, the cylinder assembly and the grille assembly. Main cabinet contains fresh air filter and has felt seals to give an air tight seal against the cylinder. Two subassemblies make up the cylinder assembly. Motor and blowers are mounted on a board that fits into the cylinder. Cylinder has closed ends and four ports. Longitudinal port is for fresh air inlet, two air outlet ports and the fourth port, covered with a filter is for recirculated and exhaust air. Cylinder assembly is suspended, in the cabinet, by a link at each end of the cylinder having the supporting holes so designed that the cylinder is held against felt seals in the main cabinet by its own weight. Grille assembly consists of steel stampings and is fastened to the main cabinet by screws. Grille stamping is made by piercing flat stock and folding it to final shape.

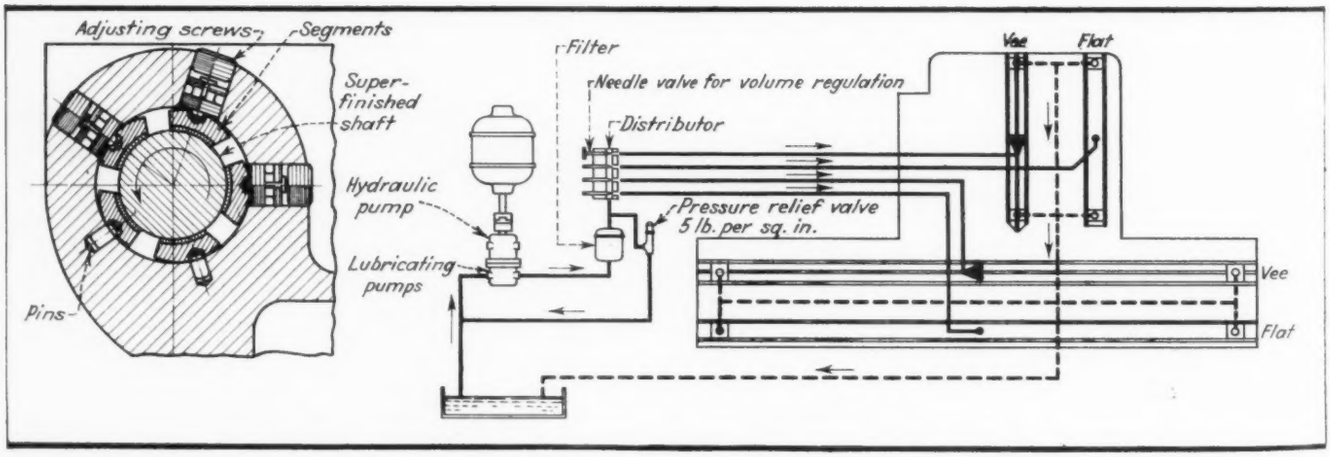


Control knob and rheostat are mounted on a bracket spot welded to the cylinder. When locknut is turned counter clockwise cylinder by its own weight rests against felt seals in the main cabinet. Clockwise rotation of the locknut lifts cylinder away from felt seals and allows a change in cylinder position.



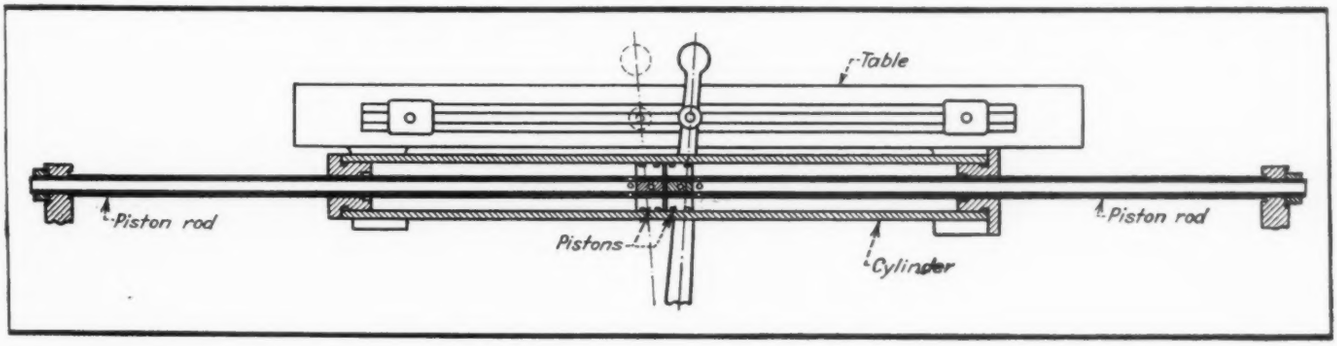
Grinding Machine Is Hydraulically Operated

HYDRAULIC TABLE TRAVERSE, cross feed mechanism, and pick feed device are incorporated into a new line of center-type grinding machines developed by Cincinnati Grinders Incorporated. Four machines are included in this line, two sizes of which were built during the war period. Headstocks for the 6 in. and 10 in. machines are powered by a 1/2 hp., a.c. motor and have four spindle speeds available through 4-step V-belt pulleys. Headstocks for the 10 in. and 14 in. machines are powered by a 1 hp., d.c. or variable voltage a.c. motor and spindle speed is changed with a rheostat. Wheel spindle drive is 15 and 7 1/2 hp. and hydraulic motors 1 1/2 and 1 hp. respectively for the large and small size machines.



Grinding wheel spindles run on Filmatic bearings placed at each end of the spindle. The Filmatic bearings consist of five segments, equally spaced around the super-finished spindle bearing diameter. Segments are located by pins and adjusting screws that allow pivoting. Spindle rotation draws oil under the segments creating a high-pressure wedge-shaped oil film. Main motor drive cannot be started until bearing compartment is completely filled with filtered oil. Pressure switch, located in bearing compartment, opens at 2 lb. per sq. in. and closes at 3 lb. per sq. in. preventing damage to the spindle should the oil supply fail.

Table and cross ways (right) are automatically lubricated through a pressure system that supplies filtered oil at about 5 lb. per sq. in. Needle valve regulates volume of oil.



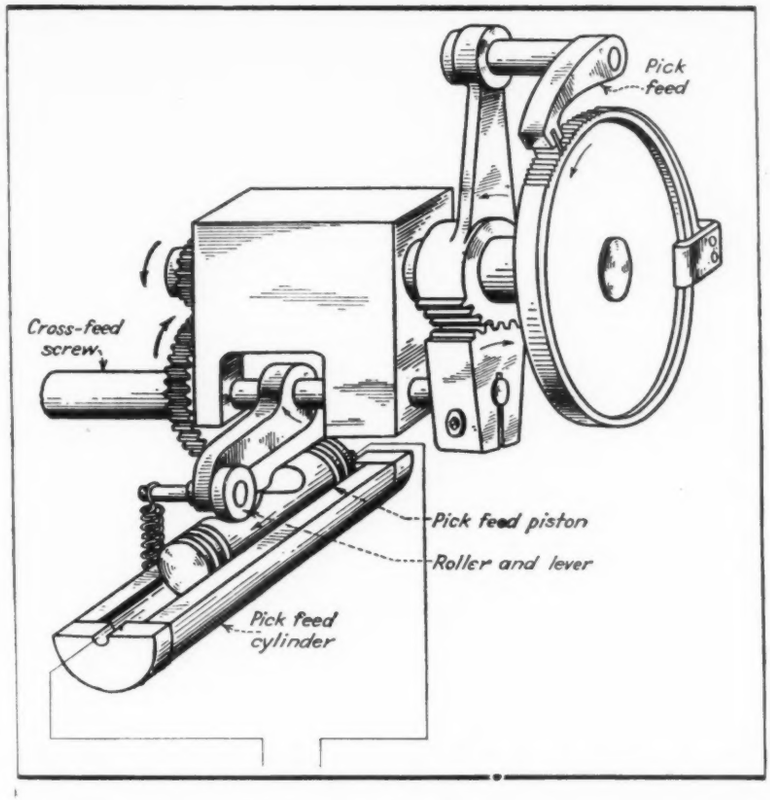
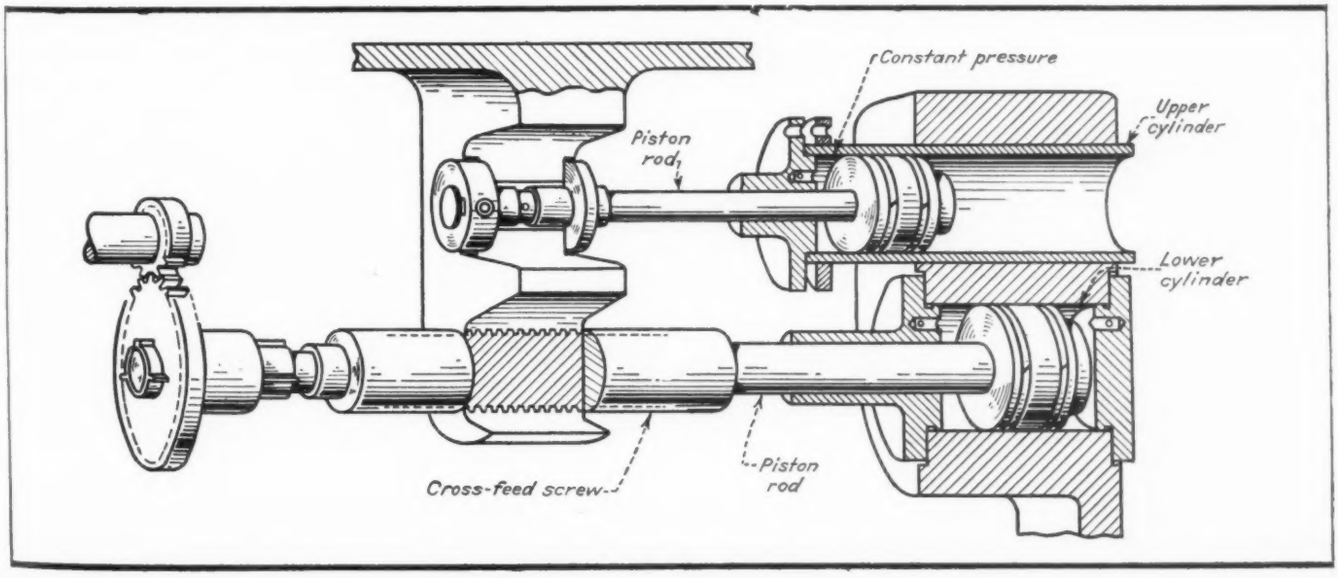


Table is traversed hydraulically by a cylinder attached to the under side (above). One end of the cylinder is fixed and the other end free to expand longitudinally. Two hollow piston rods, each connected to a separate piston, extend from the piston through the end of the cylinder. A small space between the cylinders allows the piston rods to expand longitudinally. Oil is admitted to either piston rod to traverse the table, since the pistons and rods are stationary.

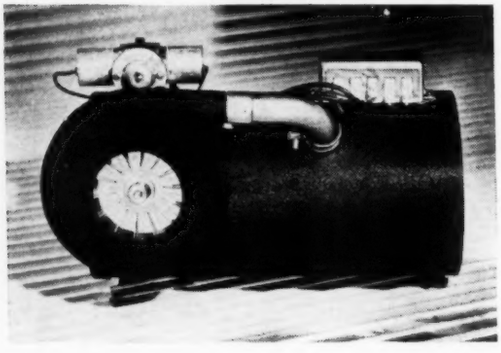
Pick feed device is hydraulically actuated (left). As table reverses a shot of oil reverses the pick feed piston that actuates the pick feed through a roller and lever.

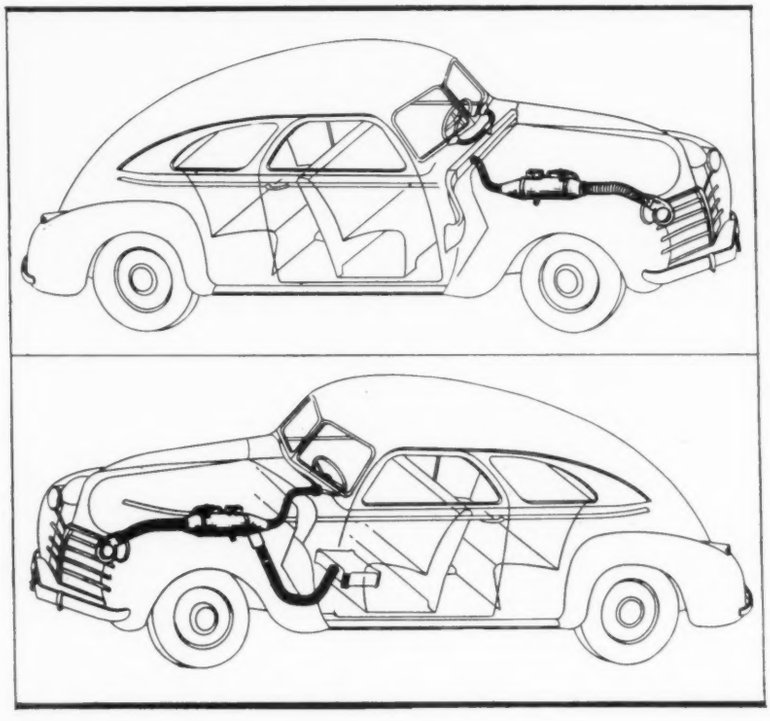
Cross-feed mechanism consists of two cylinders, (below). Upper cylinder serves as a hydraulic back-lash device. Constant oil pressure on one side of the piston compensates for backlash in the cross-feed screw. Lower cylinder is for hand and automatic servo in feed equipment. Piston rod is splined through cross-feed screw, so that screw feed can be used in the conventional manner for setting up and sizing.



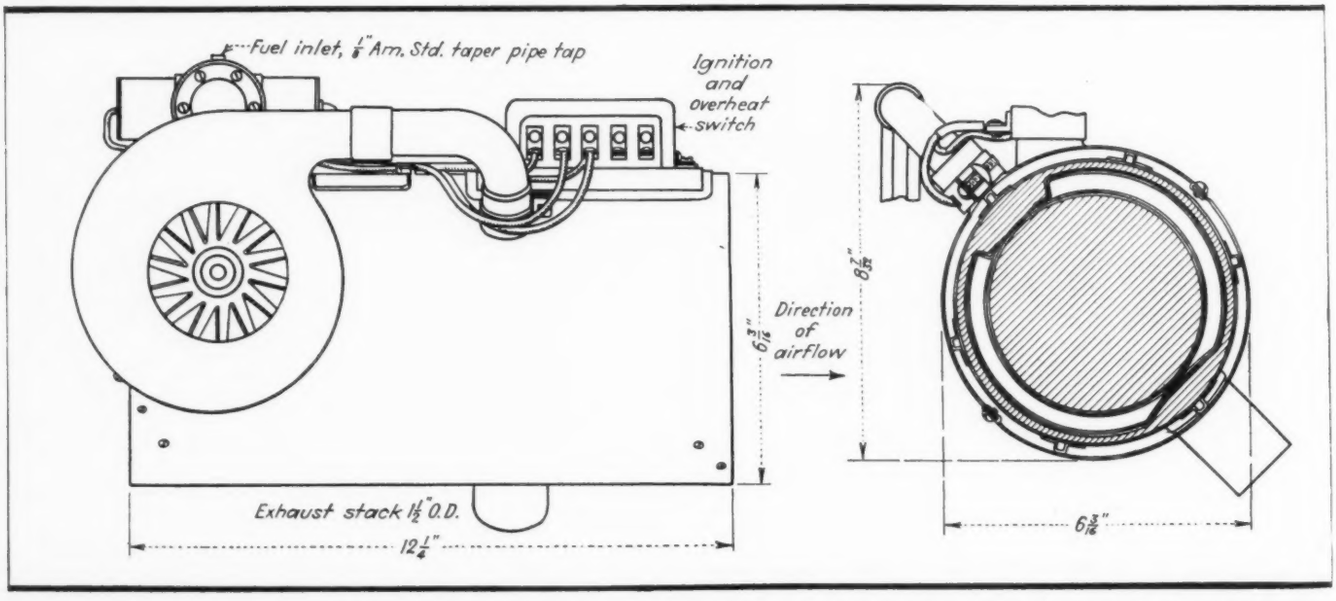
Self-Contained Automotive Heater

SMALL, compact and self-contained car heater, the Model 977 South Wind, having an output of more than 20,000 B.t.u. per hour has been developed by the South Wind Division of the Stewart-Warner Corporation. Using gasoline as fuel the heater will deliver heat in less than one minute after starting. Automatic control allows the selection of any temperature within the car and maintains this temperature in all weather. Since heater uses gasoline as a fuel it takes no heat from the engine nor does it require additional anti-freeze or increase the engine warm-up time. Heater is operated from the dash with a switch, and noise is not apparent in operation since the heater is outside the passenger compartment. Construction is all welded stainless steel.





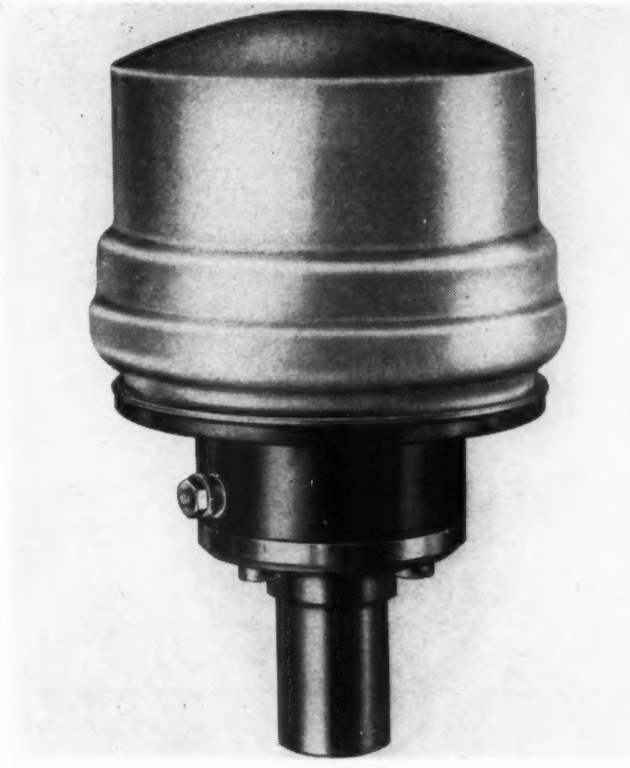
Installation is comparatively simple requiring only fuel line and electrical connections. Heater is mounted in the engine compartment and a blower draws air from behind the radiator grille. Air from the blower passes through the heater to outlets inside the car. Outlets may be placed under the seat with a separate connection to the windshield defroster or one large duct, located behind the dash, can be used to distribute the heated air to both defroster and car heater outlets.

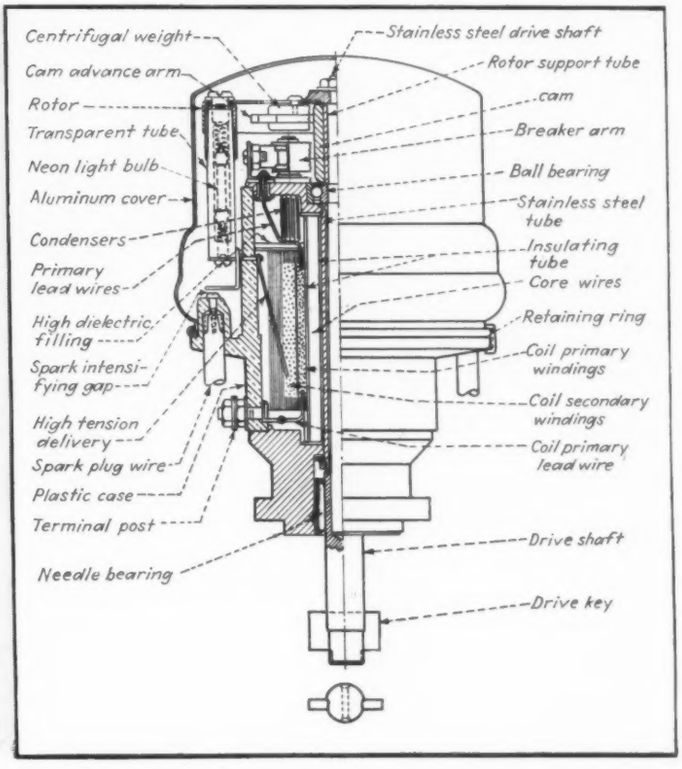


Construction of the heater is all-welded stainless steel hermetically sealed. Fan attached to the side of the heater furnishes air for combustion of the fuel. Ignition and overheat switch prevents excessive heating.

Unified Engine Ignition System

MOLDED PLASTIC case having high dielectric properties, houses a single-unit battery ignition system for aircraft, automotive, marine and industrial engines. Developed and manufactured by the LaPointe-Plascomold Corporation the coil, condenser and distributor are combined in a single case. Called Uni-power, the complete unit weighs about 2 1/2 lb. Aluminum cover protects unit against dust and moisture and provides radio shielding. Distributor rotation can be clockwise or counterclockwise. Neon light on rotor arm indicates ignition performance. Spark plug leads are attached at bottom of housing. Stainless steel is used for the drive shaft.





Random Notes

Balancing connecting rods is speeded up with a new machine developed by Snyder Tool & Engineering Co., Detroit, Mich. Formerly two distinct operations were required; weighing, then milling away excess metal. Snyder's new machine first weighs each end of the rod, then after the operator has set the overweight readings on the positive end stop hand wheels, the machine mills off the required amount of metal and unclamps the piece leaving it to hang freely on the scale rods to give the operator an automatic check on the accuracy of the operation just completed.

* * *

Paper condensers were made in wartime Germany by applying a vaporized zinc coating to the paper dielectric. Smaller by 40 per cent than conventional American foil-type condensers, the paper condensers are also 20 per cent cheaper. The paper "heals" after electrical breakdown because a thin vaporized film evaporates around the breakdown point and establishes an insulating margin. Twenty to 50 per cent higher voltages can thus be handled.

* * *

Aluminum-balsa "sandwich" is a promising new aircraft construction material developed by the Chance Vought Aircraft Company. "Metalite" consists of a high strength aluminum alloy skin bonded to a balsa core with the balsa grain perpendicular to the faces. Its strength makes the conventional stiffeners in bulkheads and web sections unnecessary.

* * *

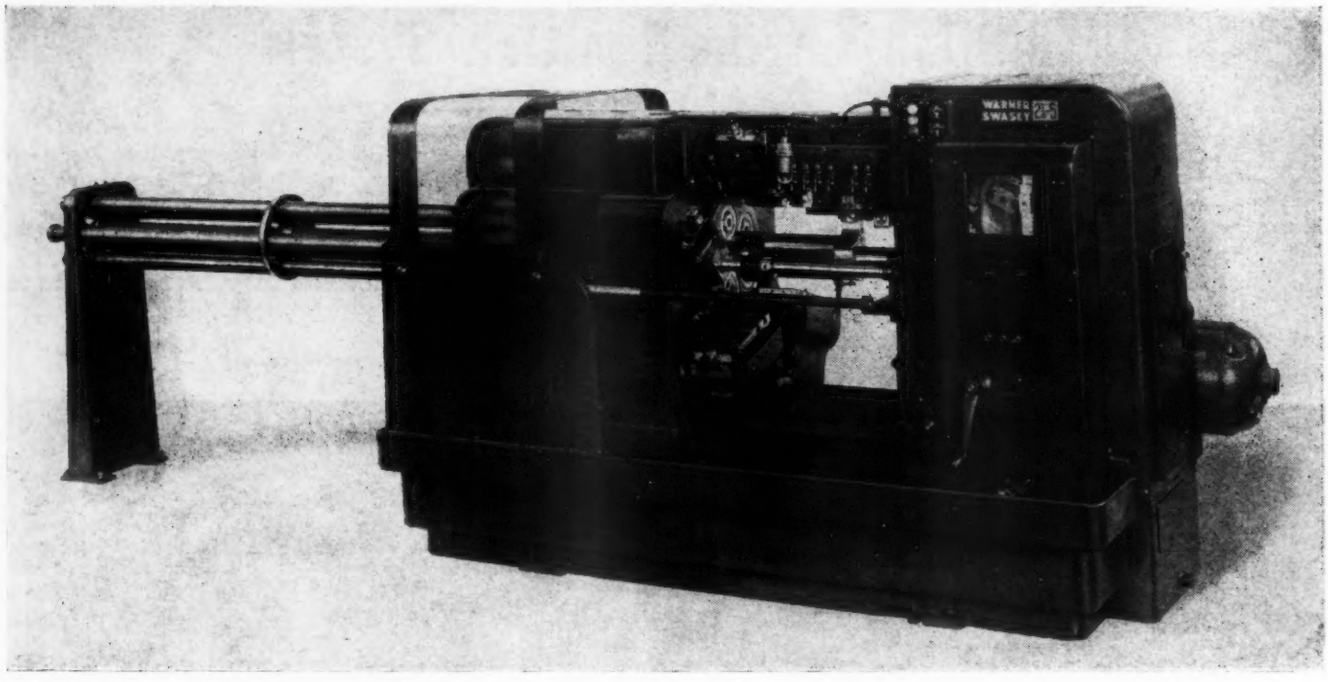
Fiber glass replaces goat skin in sensitive diaphragms for control and instrumentation equipment made by the Askania Regulator Company of Chicago. The glass diaphragms are impervious to gases and vapors, have a high resistance to alkalis and acids, and have a much longer life than goat leather.

* * *

Teflon, a new plastic by E. I. Du Pont de Nemours & Co., Inc., is reported to withstand corrosive acids to a greater degree than any known organic material.

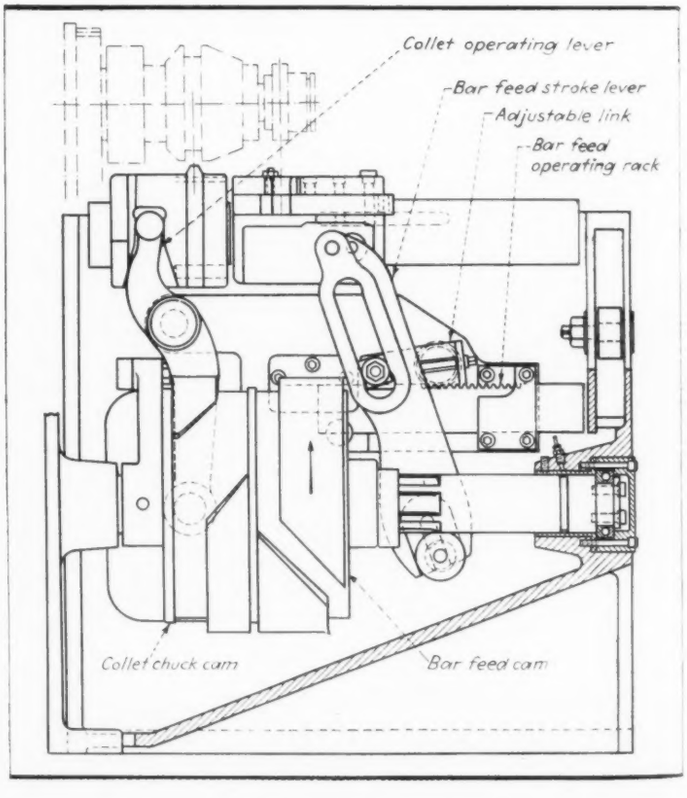
* * *

Mechanical motion is converted directly into variable electron flow with the new electron tube developed by Radio Corporation of America, Camden, N. J. The "Vibrotron" tube is a metal triode one in. long and ¼ in. dia. weighing 1/15 oz. Limited quantities are available for experimentation.

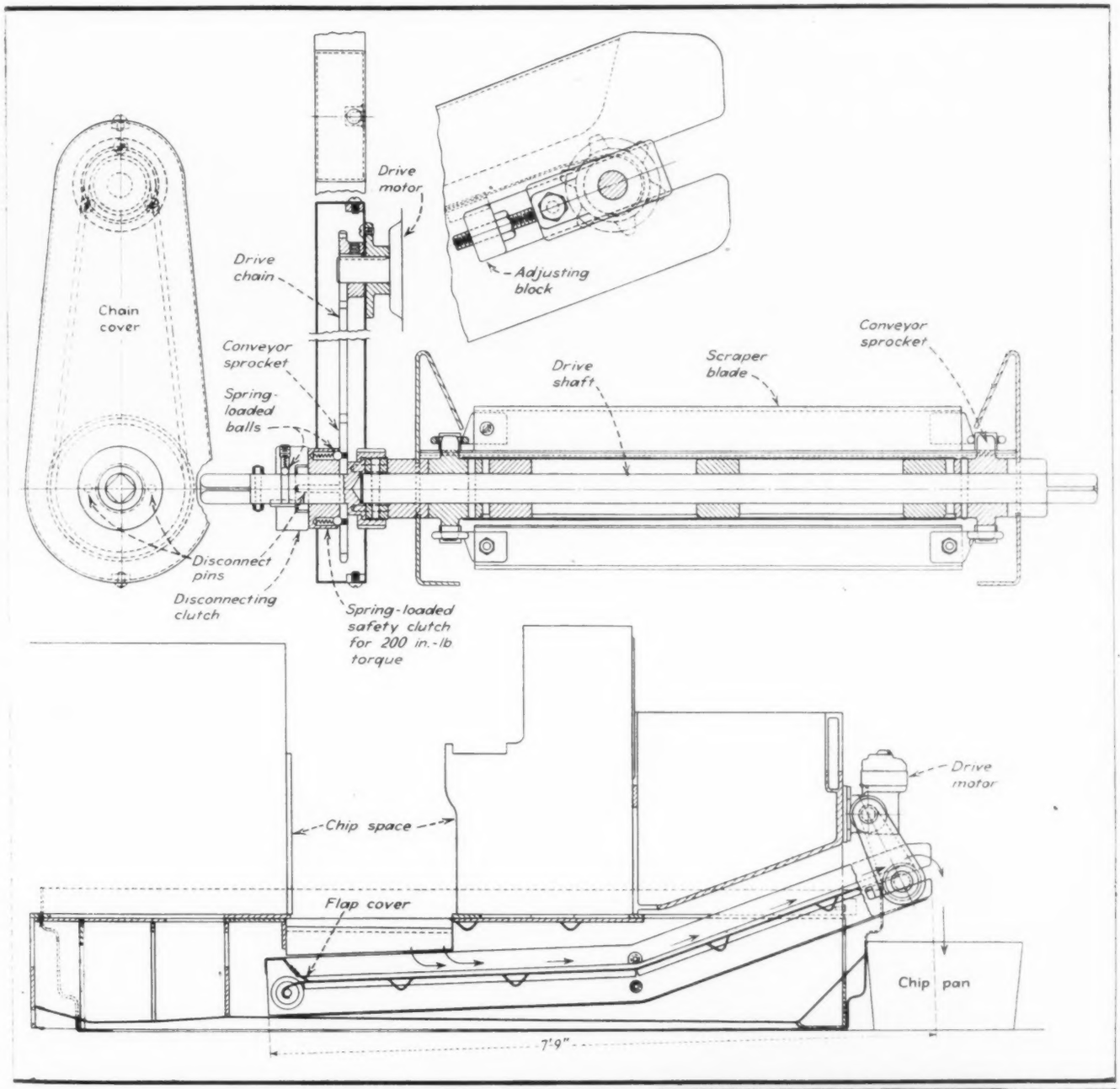


Multiple Spindle Bar Machine Eliminates Interchangeable Cams

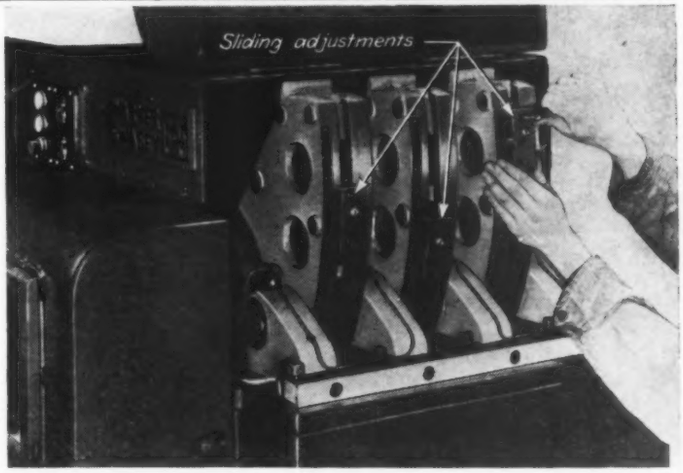
INTERCHANGEABLE CAMS are eliminated in a multiple spindle automatic bar machine manufactured by The Warner & Swasey Company. Quadrant linkages allow infinite adjustment for quick and accurate setting of the turret and the five cross-slides. Spindles are carried in a drum that indexes without shock by a mechanism not using the conventional lock bolt. Eight sets of pick-off gears and a high-low shift lever give twenty-four spindle speeds. Chips are removed by a chip conveyor equipped with a safety clutch to avoid breakage should the conveyor become jammed. Two sizes of the machine, for 1 3/4 in. and 2 1/4 in. bar stock, are available.



Bar feed has simple harmonic motion and consists of a cam operated bar feed rack operating the bar feed stroke lever through an adjustable link. Collet chuck cam is operated from the same shaft as the bar feed cam, thus giving the correct timing to the collet engagement and bar feed operations. Collet chuck tension can be checked manually to avoid danger from excessive strain.



Chip disposal conveyor is driven by a chain drive through a safety clutch consisting of nine spring-loaded balls acting against the conveyor sprocket. Connection of safety clutch to drive shaft is through a sliding disconnect clutch having two pins meshing with the safety clutch.



Length of feed strokes is mechanically controlled by a quadrant linkage combination that eliminates the need for interchangeable cams. Sliding adjustments set all feed working strokes from zero to maximum but retain the fast approach feed stroke. Cross slide strokes are controlled by adjustable linkages similar in principle to the quadrant linkage used for the feed strokes.

Clerical and Semi-Technical Services In Engineering Department Management

JAMES E. THOMPSON
Chief Production Engineer, Parkhurst and Company

Duties, control and liaison between departments, including divisions of a company remotely separated from one another. Company standards, advance bills of material, master parts lists, specifications, and handbooks are discussed in detail.

A VARIETY of services support the work of engineering designers and draftsmen. Those which require specialized technical knowledge, but do not directly pertain to product design or drafting, can be classified as technical services. Those requiring slight technical training, and those that are clerical in nature, can be considered general services.

Most technical services supply specialized assistance in solving product-design problems, but in a few organizations carry out operations of the engineering department. Engineering planning, cost control, patent research and handbook preparation are in the latter classification.

While a majority of the general services are clerical, there are some that must be considered semi-technical. Photography and parts catalogs are examples. These are considered general services because they are not mandatory sections of the engineering department. Photography is frequently part of the sales or public relations departments. Parts catalogs may be a function of the sales department—or it can be subcontracted to a company specializing in catalog preparation.

Among the more important engineering technical services are: (1) Planning; (2) cost control; (3) standards; (4) patent research; (5) loft layout; (6) materials; (7) processes; (8) specifications; (9) handbooks; and (10) illustration. Some engineering departments require additional specialized technical services, such as stress analysis and weight control. In a small engineering department one person will handle several of these services. Large departments usually have personnel assigned to each technical service.

Engineering cost control and planning have received consideration in preceding articles (See PRODUCT ENGINEERING, May 1945, p. 328-330; and June 1945, p. 367-371), and do not require further discussion.

Basic general services that are essential to the operation of an engineering department are: (1) Master parts lists; (2) parts catalogs; (3) spare parts lists; (4) photography; (5) timekeeping; (6) blueprinting; (7) drawing and print files; (8) drawing release; (9) library; (10) report release; (11) supplies; and (12) stenographic facilities and correspondence files.

Parts catalogs, spare parts lists, photography and timekeeping services can be supplied by other departments. The remainder are basic general services of the engineering department. Drawing release, drawing and print files and blue-printing have been considered in preceding articles (See PRODUCT ENGINEERING, July 1945, p. 465-470, and October 1945, p. 676-680), and consequently they are excluded from this consideration of engineering services.

The services essential to all engineering departments, and that have not been considered in preceding articles, are: Standards; advance bills of material; master parts lists; specifications; and handbooks.

Standards

Standards are established to govern drawing preparation, engineering procedures and methods, certain parts and designs. One person in engineering is designated as standards engineer to control preparation, distribution and revision of standards. All standards are carefully prepared. They eliminate the need for engineering personnel to debate the method of preparing a drawing, the procedure in specifying stock items, and so on.

It might appear that formal, published standards would only be required in large engineering departments and that small organizations could depend upon the chief engineer or chief draftsman verbally to define departmental policies, rules and procedures whenever the need arose. This is not considered desirable. Conflicting decisions may be given when long periods elapse between reoccurrence of similar questions. Too much of the executive's time may be consumed in repeating instructions on routine matters. The time spent in explaining a particular policy or procedure once is little more than that required to prepare the information as a standard. When care is exercised to issue standards only on important matters, the cost of their issuance is small.

When a provision of an engineering standard is found undesirable, or in error, a revision to correct the fault is promptly issued. Verbal deviations from a standard are not authorized, as this would lead to the entire standards system falling into disrepute. A revised standard is issued as soon as possible, and corresponding revisions made in all related standards. Where immediate change or correction is imperative, a memorandum stating the nature of the revision is issued as temporary authority for the deviation.

Standards relating to policy and procedure are furnished engineering executives and supervisors. Those relating to drafting room practices and to design and standard parts are supplied to all design and drafting personnel.

A standard part differs from other parts in that it is not identified with any particular model, but is (or can be) used universally through a range of models. Merely because a part is used on several models does not always merit its classification as a standard. To become a standard the part must be one that could conceivably be used on nearly every model.

Standard designs establish uniform methods of achieving certain manufacturing or assembly operations. Their use simplifies drafting by eliminating repetition of notes on the manufacturing drawings. Examples of one type of standard design are lightening holes, stiffener beads, rivet-installation data and electrical-cable assembly information. When these standards exist, the drawings show only locating dimensions for lightening holes, beads, and the like, with notes such as "SD1001-6 LIGHTENING HOLE," "SD-1072-3 BEAD" and "INSTALL RIVETS PER SD1010". The related standards drawings provide detail fabricating information for the manufacturing departments.

Another type of standard design drawing shows "envelope" dimensions for design

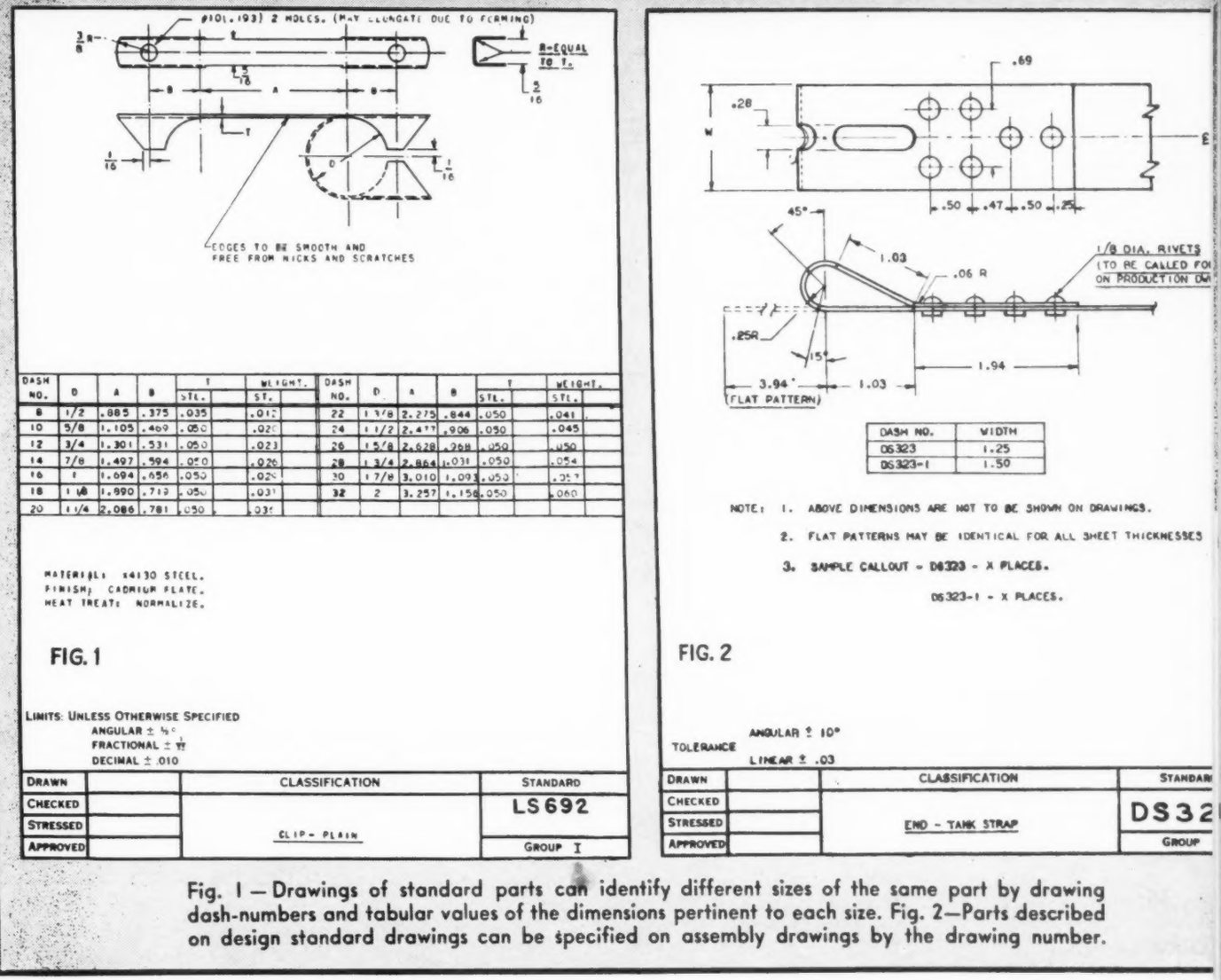

Ø10L (.193) 2 HOLES. (MAY ELONGATE DUE TO FORMING)
B-EQUAL TO T.
EDGES TO BE SMOOTH AND FREE FROM NICKS AND SCRATCHES
DASH NO. | D | A | B | T STL. | WEIGHT ST. | DASH NO. | D | A | B | T STL. | WEIGHT STL.
8 | 1/2 | .885 | .375 | .035 | .012 | 22 | 1 3/8 | 2.275 | .844 | .050 | .041
10 | 5/8 | 1.105 | .469 | .050 | .020 | 24 | 1 1/2 | 2.477 | .906 | .050 | .045
12 | 3/4 | 1.301 | .531 | .050 | .023 | 26 | 1 5/8 | 2.628 | .968 | .050 | .050
14 | 7/8 | 1.497 | .594 | .050 | .026 | 28 | 1 3/4 | 2.864 | 1.031 | .050 | .054
16 | 1 | 1.694 | .656 | .050 | .029 | 30 | 1 7/8 | 3.010 | 1.093 | .050 | .057
18 | 1 1/8 | 1.890 | .719 | .050 | .031 | 32 | 2 | 3.257 | 1.156 | .050 | .060
20 | 1 1/4 | 2.086 | .781 | .050 | .035
MATERIAL: x4130 STEEL.
FINISH: CADMIUM PLATE.
HEAT TREAT: NORMALIZE.
FIG. 1
LIMITS: UNLESS OTHERWISE SPECIFIED
ANGULAR ± ½°
FRACTIONAL ± 1/32
DECIMAL ± .010
DRAWN | CLASSIFICATION | STANDARD
CHECKED | CLIP - PLAIN | LS692
STRESSED | | GROUP I
APPROVED |
.69
.28
W
.50 .47 .50 .25
45°
1.03
.06 R
1/8 DIA. RIVETS
.25R
15°
1.94
3.94° (FLAT PATTERN)
1.03
DASH NO. | WIDTH
DS323 | 1.25
DS323-1 | 1.50
NOTE: 1. ABOVE DIMENSIONS ARE NOT TO BE SHOWN ON DRAWINGS.
2. FLAT PATTERNS MAY BE IDENTICAL FOR ALL SHEET THICKNESSES
3. SAMPLE CALLOUT - DS323 - X PLACES.
DS323-1 - X PLACES.
FIG. 2
TOLERANCE ANGULAR ± 10°
LINEAR ± .03
DRAWN | CLASSIFICATION | STANDARD
CHECKED | END - TANK STRAP | DS32
STRESSED | | GROUP
APPROVED |

Fig. 1 — Drawings of standard parts can identify different sizes of the same part by drawing dash-numbers and tabular values of the dimensions pertinent to each size. Fig. 2—Parts described on design standard drawings can be specified on assembly drawings by the drawing number.

components standardized by a recognized committee or governmental agency. Examples of these are SAE standard mounting provisions for automotive and aircraft accessories, and Army-Navy dimensional standards for aircraft accessories.

During the preliminary design of each new product a careful study is made of existing standard parts, and as many of them used as possible. Hardly a day passes without at least one use being found for a standard part where a special part would otherwise have been designed. The application of design standards is encouraged to effect the savings in design and drafting time accruing from their use.

All changes in company standards require approval of the standards engineer. They become effective on future production, it being understood that parts on hand at the time of the change will be used until the stock is exhausted. When a change will affect interchangeability of the standard part, a new standard under a new drawing number is issued to describe the "changed" part. The previous version of the part is retained in the handbook, marked with the notation "Inactive for design, use (new standard)".

When a company has several operating divisions there is introduced the problem of coordinating standards between divisions. The standards engineer at the company's home-office then becomes standards coordinator, and a uniform company standards handbook is used by all divisions. This avoids duplication of standards, and insures that all existing standards are available to each division.

Materials

Selection of the proper material for a given part is an important consideration in every design. It is necessary to determine the most economical material to perform the required function, without compromising safety and service life. The material selected must be one that can be obtained in the required raw-stock form in time to meet production schedules. This technical service is sufficiently important to justify assigning the function of "materials engineer" to one person. In a small department this can be made a responsibility of the standards engineer, while larger engineering offices will require the full-time services of a competent person adequately to care for materials problems.

The materials engineer maintains an index of approved materials that are listed for use on manufacturing drawings. Usage of other materials by design and drafting personnel requires approval of the materials engineer. This insures that the proper items will be specified on drawings, and avoids both procurement of unnecessarily expensive items, and the use of dubious products that may lead to service difficulties.

One of the problems encountered, during the development of a new design or a major change, is ordering the required material early enough to insure its being in stock when required for construction. This problem confronts not only engineering departments engaged in product design, but also those at processing plants where the material is machined and fabricated. Substitutions and changes in materials due to the development of a new product also require cooperation between designers and materials engineer.

Consider the engineering department of a cement manufacturing plant. Their work is principally concerned with the construction, installation, modification and repair of processing equipment. Large inventories of raw-stock and purchased parts are maintained for maintenance purposes. Advance information on changes affecting this inventory will not only permit ordering material for delivery in advance of actual need, but will also

Specimen Page from Materials Index, Which Is Used by Designers and Draftsmen in Specifying Materials for Parts

Material	Specification	Uses and Remarks
BRASS Wire	Fed-QQ-W-321 (E) Grade A—Annealed, Soft Grade B—As Drawn Grade C—Spring Temper	Specify "Grade A" for beading, seizing, "Grade B" for hinge-pins, rivets, lamp guards; "Grade C" for springs, lock rings.
BRONZE Bar, Rod, Shapes—Aluminum-Bronze	Fed-QQ-B-666 (E) Grade B Type I —Rod Type II —Bar Type III—Shapes	Good bearing and corrosion-resistant qualities.
Bar, Rod, Shapes—Manganese-Bronze	Fed-QQ-B-721 (E)	Refer to spec. for classes and tempers.
Bar, Rod, Shapes—Phosphor-Bronze	Fed-QQ-B-746 (E) Grade A Type I —Rod Type II —Bar Type III—Shapes	Highly stressed bolts, pump rods; Specify "Hard". Electrical contacts, springs.
Cable, Extra Flex. 7 x 19	Navy 22C5	1/8-in. Dia. and up.
Castings—Aluminum-Bronze	AN-QQ-B-672	For centrifugal casting, highly loaded bearings. Slightly harder than Fed-QQ-B-671.

Fig. 3—When this form is used for products other than aircraft the word "unit" can be substituted for the words "plane" and "airplane".

Fig. 4—Master Parts List of the type shown here can be prepared on 11x17 in. printed vellum forms.

# DENOTES SEPARATE PARTS LIST

LAST CORR	Item No.	ZONE	PART NUMBER	Change Letter	REQ'S L.H.	REQ'S R.H.	SET BACK 1	2	3	4	
	1		20002	E	1						Valve Assem - Sequence
	2		99114	-	1						Decal - Nameplate
	3		99100-33	A	1						Nameplate
	4		AN535-00-2	-	2						Screw - Drive
	5		AF995-32-4	-	2						Lockwire
	6		AF500A10-6	-	6						Screw
	7		20133	A	1						Plate
	8		20127	-	1						Plug - Check
	9		AN6227-11	-	1						Packing
	10		20107	-	1						Spring - Check
	11		W508-8D	-	1						Check Assem
	12		W510-8D-4	B	1						Poppet
	13		W510-8D-5	A	1						Sleeve
	14		AN6227-7	-	1						Packing
	15		20126	A	1						Plug - Poppet
	16		AN6227-14	-	1						Packing
	17		20105	B	1						Spring - Poppet
	46										
	47										

DATE ISSUED	PARTS LISTER	CHECKER	MODEL
7/18/45	D. Johnson	J. E. Thompson	202
SUPERSEDES			
6/28/45			
M.P.L. CHG. LTR.	FIG. 4		
A			

REPLACES DRAWING NO. | VALVE ASSEMBLY - SEQUENCE

AEMO Nº 0001

ITEM	REQUIRED PER PLANE	REQUIRED SPECIAL	SIZE	DESCRIPTION	SPECIFICATION	CATALOG REFERENCE, VENDOR, ETC.	PART NO.	REQUISITION NUMBER P. O. NUMBER	REQUIRED DELIVERY DATE
1									
2									
17									

ADVANCE ENGINEERING MATERIAL ORDER

PREPARED BY / DATE — REMARKS — SHEET OF
CHECKED BY / DATE — ISSUED
APPROVED BY / DATE — REVISION OF A.E.M.O.
PROJECT
MODEL
DWG. NO. — GROUP — INSTALLATION OR ASSEMBLY TITLE — AIRPLANE SERIAL NO.
AEMO - Nº 0001

COPY 4—To Release Group

FIG. 3

avoid accumulating large inventories of materials and parts soon to be rendered obsolete by pending changes.

Early ordering of materials for construction purposes, and notification of materials changes is made by an "advance engineering material order" (AEMO), using the form shown at Fig. 3, and issued by the materials engineer. An AEMO is issued as soon as the design has reached the stage where a majority of the materials required can be accurately estimated. Copies are forwarded to the factory material control group. Material control investigates the availability of the material, and either orders the required material or allocates it from stock.

All materials and parts, including castings, standard parts, nuts, rivets and the like, required for the job are listed on the AEMO. Revised AEMOs are issued if made necessary by design changes or additions, and to authorize material substitutions. They show the date of revision and the number of the original AEMO in the spaces provided. It is the duty of all engineering supervisors to notify the materials engineer of changes in material requirements. Revised AEMOs are not issued after release of the drawing for which the AEMO is prepared, as alterations can then be recorded as drawing changes (See PRODUCT ENGINEERING, January 1946, and March 1946.

Master Parts List

The master parts list (MPL) shows all items required to construct a particular article or model, in the approximate order of their assembly, and lists the raw-stock material required to manufacture each part. It is an invaluable guide for all departments concerned with manufacturing the parts described by engineering drawings. It is the most important document issued by the engineering department.

It is customary to prepare a separate MPL for each assembly drawing when large, complex articles are to be manufactured. When a comparatively simple article comprising a small number of assemblies is concerned, the complete article can be listed on a single MPL, Fig. 4, which is prepared to accompany the final assembly drawing.

Issuance and maintenance of the master parts lists is assigned to a capable person, who is provided with ample facilities. A properly prepared, accurate MPL is a valuable asset to every department concerned with manufacture of parts described by engineering drawings, incorrect MPLs are worse than useless.

Master parts lists are prepared and issued concurrently with drawing releases. They eliminate the need for drawing bills of materials, as these would duplicate a considerable portion of the MPL information. Concurrent release is achieved by establishing

MASTER PARTS LIST

PART NAME	EFFECTIVE ON FROM	EFFECTIVE ON THROUGH	SIZE	MATERIAL DESCRIPTION	MATERIAL SPEC.	SPECIAL INFORMATION
	202-1 &	Up	–	–	–	
	202-1	202-2474		Purchased part		LA Stamp & Stationery
	202-2475&	Up		Purchased part		LA Stamp & Stationery
	202-2475&	Up		Purchased part		Standard
	202-1 &	Up		Purchased part		Standard
				Purchased part		Standard
			1-3/4x2-3/4	1/8 24ST strip	AN-A-12,Con T	
			3/4	13/16D 24ST rod	QQ-A-354,Con T	
				Purchased part		Standard
			25-1/2	.033D Music wire	AN-W-17	
				Purchased part		J.A. Whitney Corp.
				Purchased part		J.A. Whitney Corp.
				Purchased part		J.A. Whitney Corp.
				Purchased part		Standard
			13/16	1D 24ST rod	QQ-A-354,Con T	
	202-1 &	Up		Purchased part		
	202-1	202-2110	8-1/2	.063D Music wire	AN-	

NEXT ASSEMBLY	REQS. PER UNIT	REQS. PER ASSEM.	EFFECTIVE ON FROM	EFFECTIVE ON THROUGH	REMARKS
-16365	2	1	202-1	& UP	1. 17ST Bar or Rod, Spec FED-QQ-A-351 may be substituted for 24ST Bar or Rod, Spec FED-QQ-A-354.

DRAWING TITLE

PAGE 1 OF 1 PAGES | JOB NUMBER | CHG. LETTER E | DRAWING NO. 20002

the MPL function as a station in the release system. New and revised MPLs are prepared to accompany each drawing release. The information required for the MPLs is obtained from the drawings and from notices of change.

The compilation of MPLs begins with the release of the first drawing for a new article and continues through its production. A master parts-list index is provided for MPLs describing large, complex articles. This index lists and assigns reference numbers to the major assembly drawings comprising the final article, arranged in logical functional groupings. Each MPL is identified by its proper reference number for ease of filing in the correct assembly order.

Complete master parts lists, comprising the current issue of each MPL, are given to all departments. These are periodically revised by the MPL group, as experience indicates that individual recipients of the MPL cannot be depended upon properly to place new and revised pages in their correct locations. The MPL for each model begins with an index and the final assembly drawing, followed by sections describing each major assembly called-out on the final assembly. Each section begins with the MPL for a major assembly. This is followed by MPLs for the assemblies and subassemblies that make up the major assembly, arranged in the approximate order used by the factory to assemble the article.

Preparation of specifications pertaining to company products is assigned to a member of the engineering department designated as the "specifications engineer". This assignment includes proposal, sales, construction, and process specifications; design information reports; and other work relating to the presentation of engineering information in a manner that is both interesting and accurate. Some companies place the specification engineer in other departments.

The position combines the talents of engineer, journalist and commercial artist. In a small engineering department the function of specifications engineer may be combined with the duties of the standards engineer.

The specifications engineer begins work before the first line of a new design is placed upon a drawing. A design-information report giving the broad features and aims of the design is prepared as a guide for the engineering department. When the new design takes definite form, a preliminary specification is prepared to serve as a basis for mutual consideration of the design by all parties concerned. When all details of the design have been agreed upon and developed, a detail specification is prepared as the basis for final discussion and contract between manufacturer and customer.

As construction of the first article of the new design progresses, a construction specification is prepared. This serves as a permanent, illustrated history of the constructional features of the design. It will be used by the engineering department in future developments, and by the customer when manufacturing rights are involved. Throughout the development of the design, until the last manufacturing drawing is completed, the specifications engineer keeps the design-information report up-to-date, so that everyone concerned may be constantly informed of current design requirements.

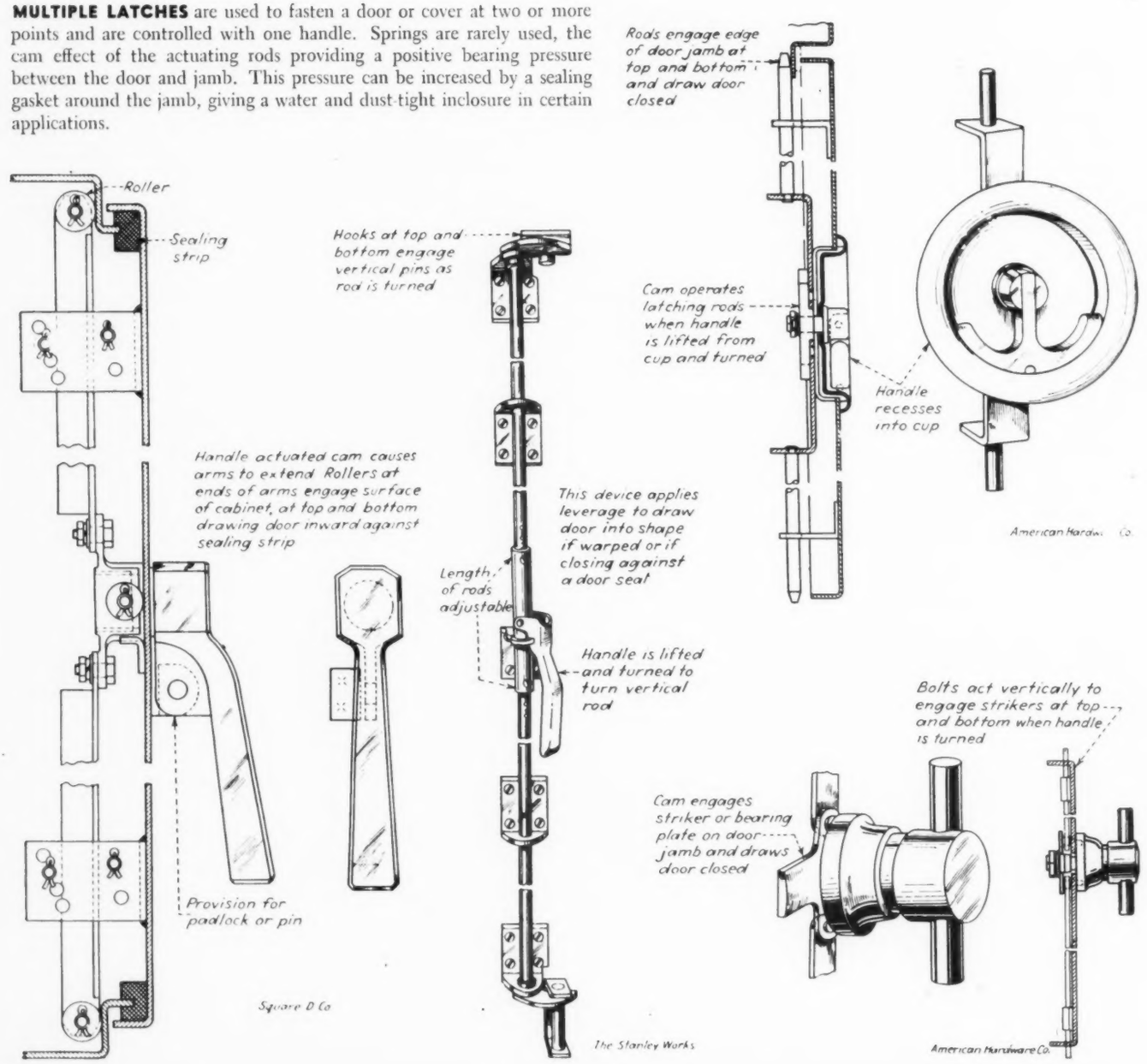
All engineering reports, irrespective of their origin, are routed to the specifications engineer for final editing. In this manner all documents coming from the engineering department maintain a consistent, uniform style. A majority of engineers are poor journalists, and editing all reports by the specifications engineer insures correctness of outline, pagination, paragraphing, grammar, and composition.

The specifications engineer in a small engineering department can prepare all handbooks of instruction relating to company products. A large engineering department will have personnel permanently assigned to handbook and service engineering functions. A high standard of accuracy is maintained for all handbooks, as customer satisfaction is greatly dependent upon accurate and practical operational maintenance and repair instructions. All handbooks are carefully checked prior to their release—by both engineering and factory personnel—to insure that the information is not only technically correct but also practical of accomplishment by the average user or maintenance mechanic.

—Manual and Automatic Latching Devices

Methods of door and cover latching range from simple gravity catches to elaborate cam and roller mechanisms. The principle of operation of many latches is fundamentally the same, regardless of whether the latch is designed as a finished hardware fitting or as a unit for industrial applications. Various latches are designed for one type of application and their characteristics prohibit interchanging from one type of application to another. Some latches classified as multiple, friction, and hook types are illustrated. Each classification is sub-divided according to methods of mounting, actuation, operation, and use.

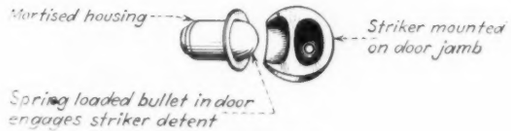
MULTIPLE LATCHES are used to fasten a door or cover at two or more points and are controlled with one handle. Springs are rarely used, the cam effect of the actuating rods providing a positive bearing pressure between the door and jamb. This pressure can be increased by a sealing gasket around the jamb, giving a water and dust-tight inclosure in certain applications.


Rods engage edge of door jamb at top and bottom and draw door closed
Roller
Sealing strip
Hooks at top and bottom engage vertical pins as rod is turned
Cam operates latching rods when handle is lifted from cup and turned
Handle recesses into cup
American Hardware Co.
Handle actuated cam causes arms to extend. Rollers at ends of arms engage surface of cabinet, at top and bottom drawing door inward against sealing strip
This device applies leverage to draw door into shape if warped or if closing against a door seat
Length of rods adjustable
Handle is lifted and turned to turn vertical rod
Bolts act vertically to engage strikers at top and bottom when handle is turned
Cam engages striker or bearing plate on door jamb and draws door closed
Provision for padlock or pin
Square D Co
The Stanley Works
American Hardware Co.

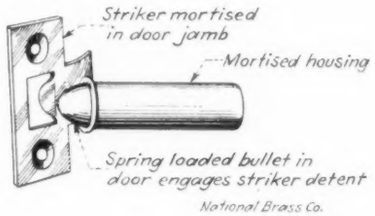
For Doors, Covers and Lids—I

FRICTION LATCHES utilize spring pressure in their actuation. The springs can be of helical, clip, or cantilever design and their resiliency controls the effectiveness of the latch. Friction latches are either mortised or surface-mounted, depending upon the particular application and in either instance are generally invisible when the door or cover is closed.

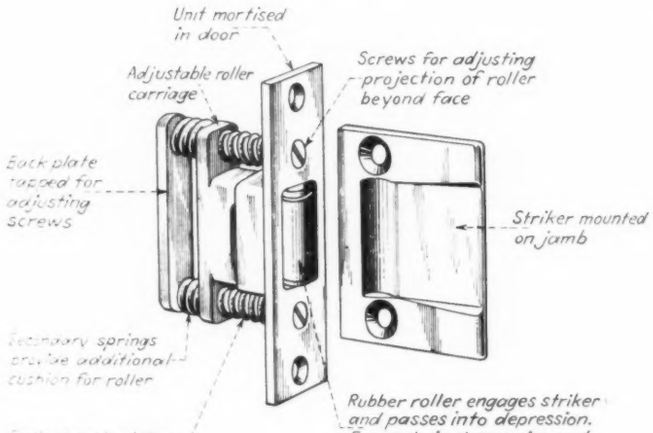
MORTISED FRICTION TYPES

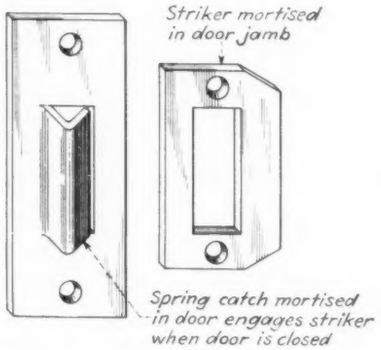


National Brass Co.



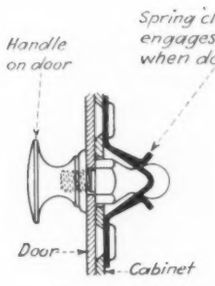
National Brass Co.


Unit mortised in door
Adjustable roller carriage
Screws for adjusting projection of roller beyond face
Back plate tapped for adjusting screws
Striker mounted on jamb
Secondary springs provide additional cushion for roller
Springs restraining roller carriage from face
Rubber roller engages striker and passes into depression. Expands in depression and holds door securely
Glynn-Johnson Corp.

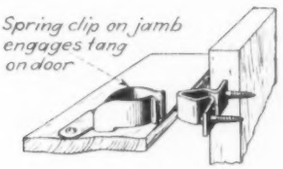


National Brass Co.

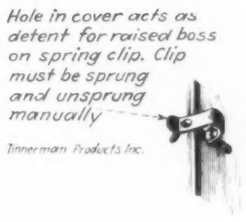
SURFACE FRICTION TYPES



Tinnerman Products Inc.



Lawrence Bros. Inc.



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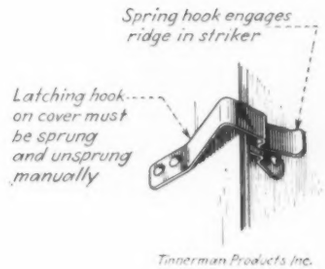

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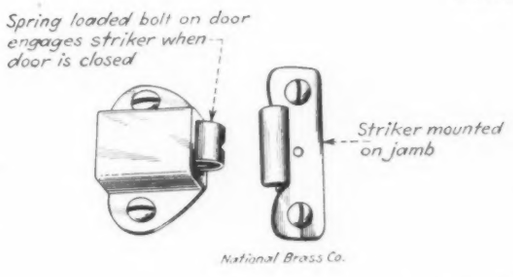
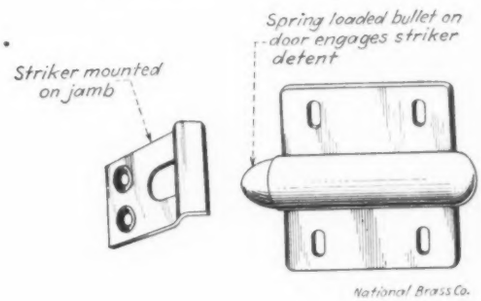
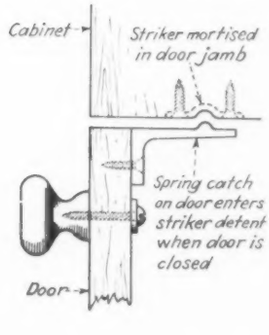


National Brass Co.



National Brass Co.

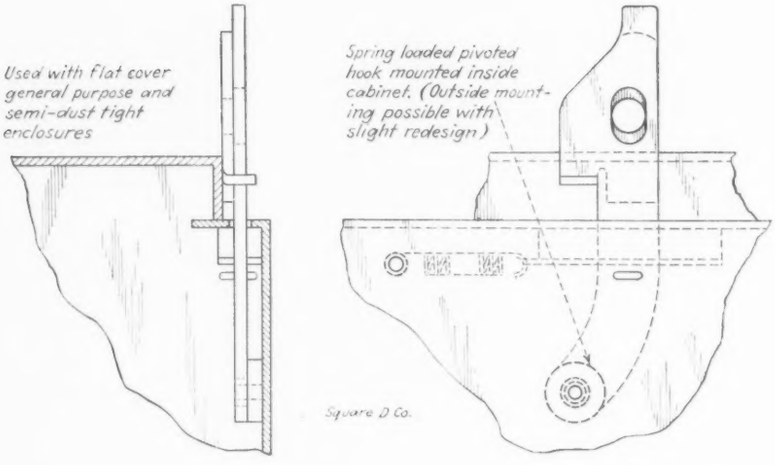


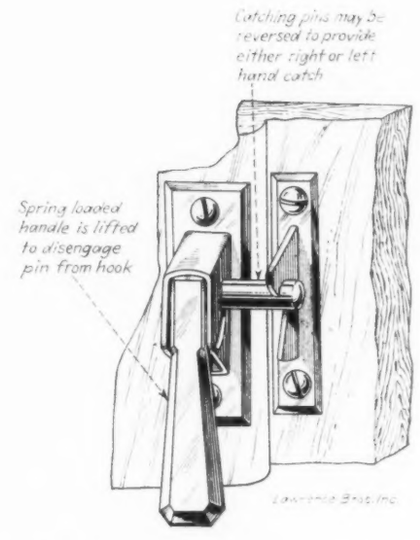
(Continued on next page)

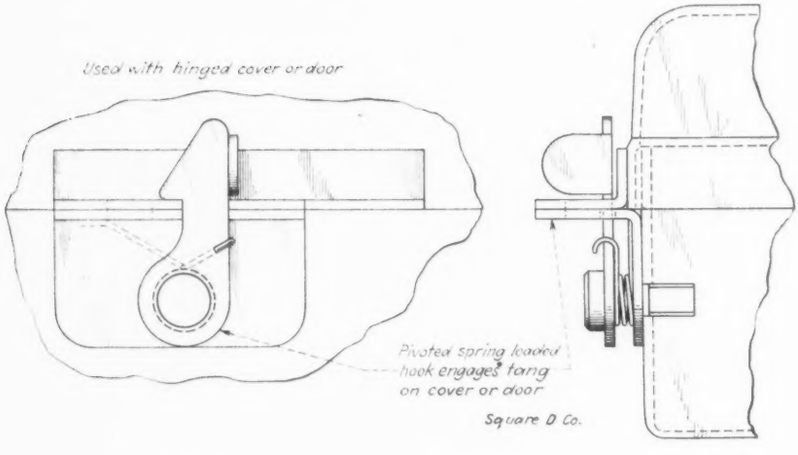
LATCHING DEVICES (continued)

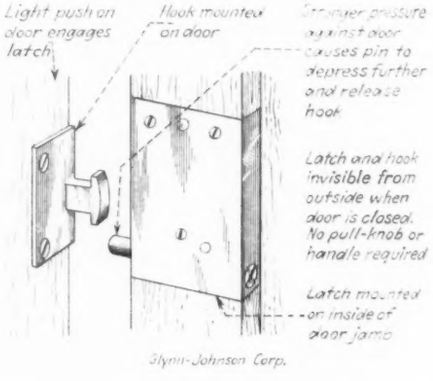
HOOK LATCHES for doors or covers are of either the gravity or spring type, both of which usually require manual latching and unlatching. In the gravity type, the hook falls into place when aligned with its mating element. In the spring type the hooking action is controlled and under pressure from the actuating spring until manually unlatched.

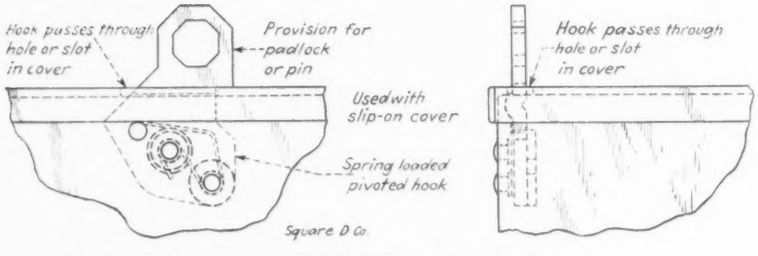
SPRING HOOK TYPES

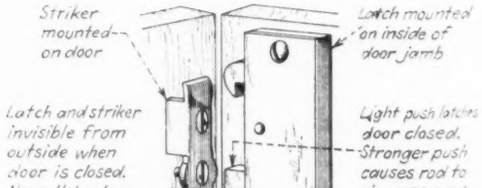

Used with flat cover general purpose and semi-dust tight enclosures
Spring loaded pivoted hook mounted inside cabinet. (Outside mounting possible with slight redesign)
Square D Co.

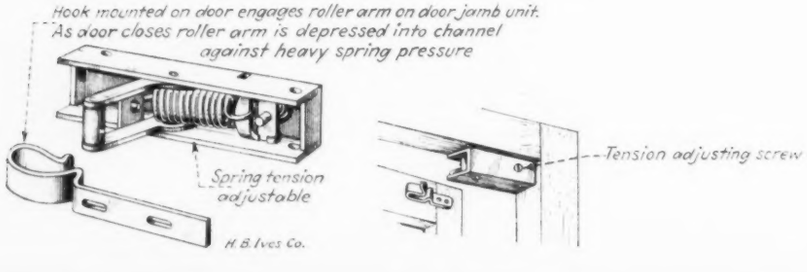

Catching pins may be reversed to provide either right or left hand catch
Spring loaded handle is lifted to disengage pin from hook
Lawrence Bros. Inc.

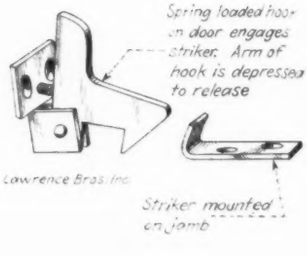


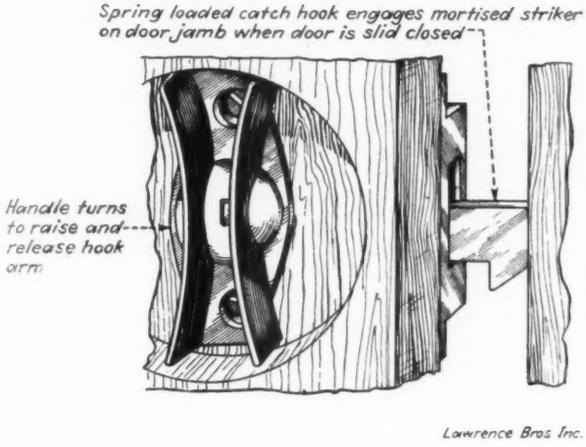

Light push on door engages latch
Hook mounted on door
Stronger pressure against door causes pin to depress further and release hook
Latch and hook invisible from outside when door is closed. No pull-knob or handle required
Latch mounted on inside of door jamb
Glynn-Johnson Corp.


Hook passes through hole or slot in cover
Provision for padlock or pin
Used with slip-on cover
Spring loaded pivoted hook
Square D Co.
Hook passes through hole or slot in cover

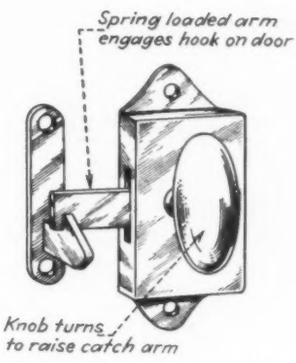

Striker mounted on door
Latch mounted on inside of door jamb
Latch and striker invisible from outside when door is closed. No pull-knob or handle required
Light push latches door closed. Stronger push causes rod to depress and release catch
H.B. Ives Co.


Hook mounted on door engages roller arm on door jamb unit. As door closes roller arm is depressed into channel against heavy spring pressure
Spring tension adjustable
H.B. Ives Co.
Tension adjusting screw

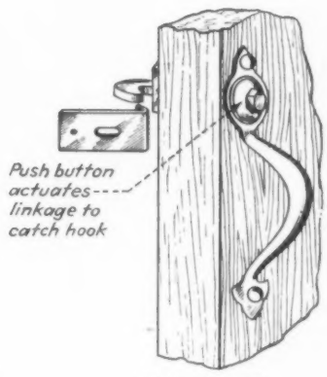

Spring loaded hook on door engages striker. Arm of hook is depressed to release
Lawrence Bros. Inc.
Striker mounted on jamb


Spring loaded catch hook engages mortised striker on door jamb when door is slid closed
Handle turns to raise and release hook arm

Lawrence Bros. Inc.

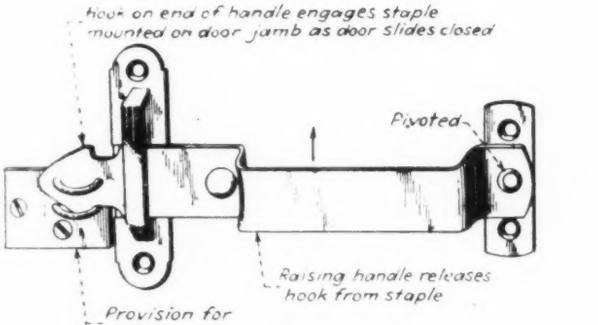


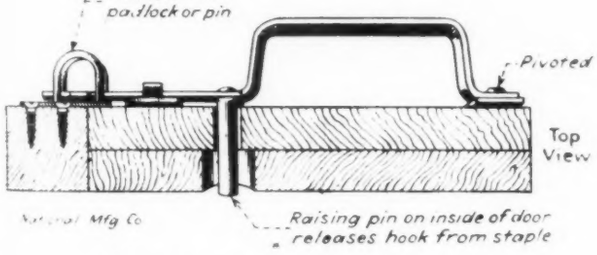
National Brass Co.



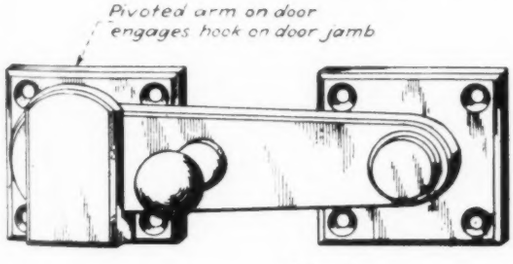
National Brass Co.

GRAVITY HOOK TYPES

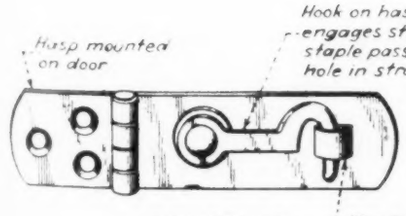

hook on end of handle engages staple mounted on door jamb as door slides closed
Pivoted
Raising handle releases hook from staple
Provision for padlock or pin



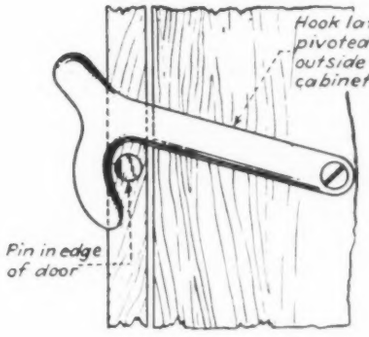
National Mfg. Co.

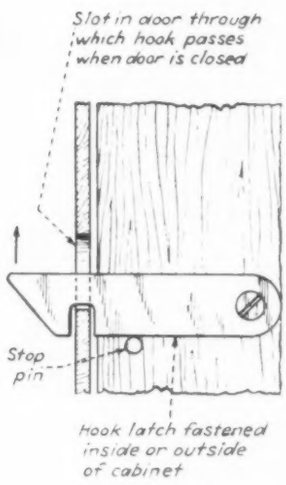


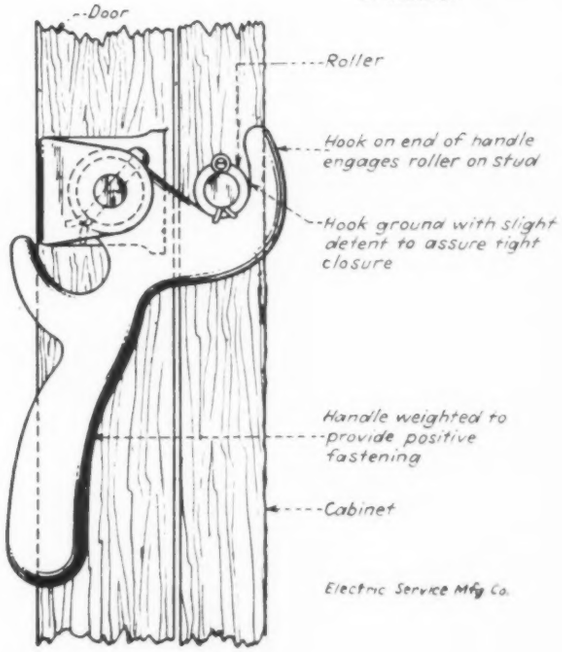
H. B. Ives Co.


Hasp mounted on door
Hook on hasp strap engages staple as staple passes through hole in strap
Staple mounted on door jamb

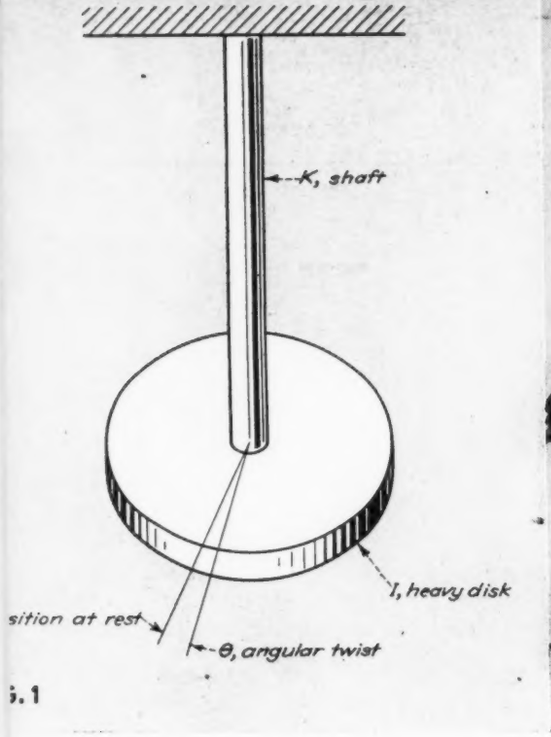
National Brass Co.






Door
Roller
Hook on end of handle engages roller on stud
Hook ground with slight detent to assure tight closure
Handle weighted to provide positive fastening
Cabinet

Electric Service Mfg. Co.



5.1

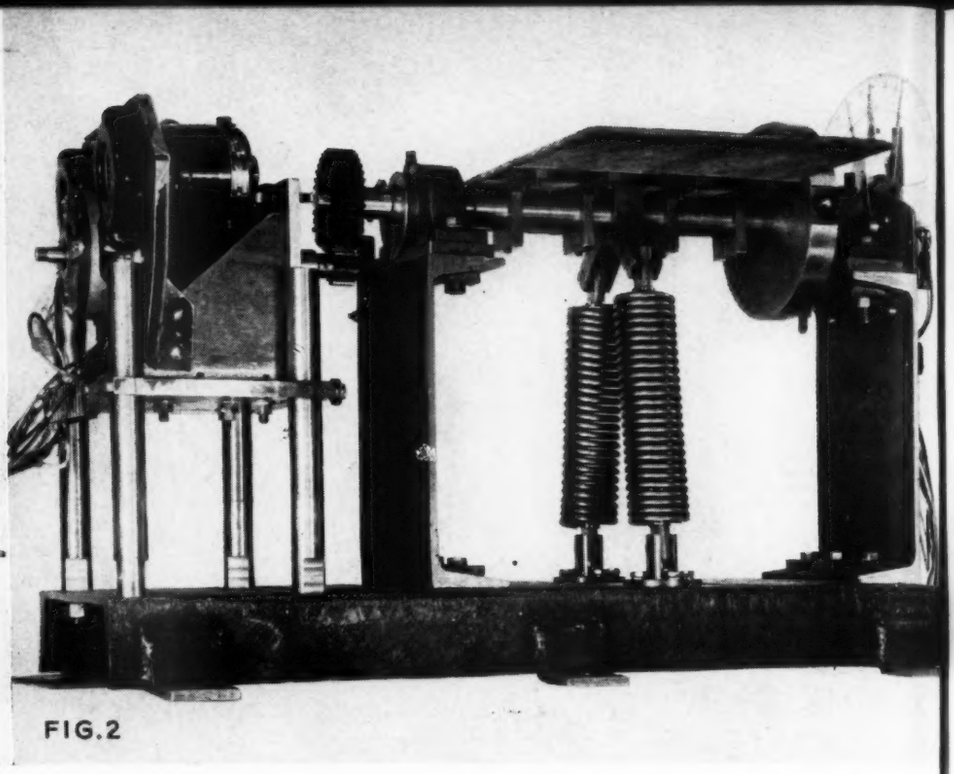


FIG.2

Fig. 1—Torsional oscillating system consisting of a shaft and a heavy disk. Fig. 2—On the mechanical mock-up a board approximates the windage of the radar antenna and a steel disk has the equivalent antenna inertia. Resilience characteristics of shaft in Fig. 1 are replaced in the mock-up by a free shaft in bearings restrained by two tension springs.

Torsional Oscillating Motion With Self-Synchronized Impulse

DAVID B. NICHINSON
Project Engineer, Kollsman Instrument Division, Square D Company

A new mechanical approach to the problem of maintaining an oscillation at a specified frequency and a variable amplitude in a resonant system. The principles and method discussed can be applied advantageously to obtain a smooth rapid oscillation with a minimum of power and power transmitting parts.

A MECHANICAL SYSTEM vibrating at its natural frequency requires but a minimum of power to maintain oscillation. All accelerations are accomplished by energy stored within the system, while power need be introduced only to compensate that dissipated by friction forces. A commonplace example illustrating the small amount of power necessary to maintain oscillations of a resonant system is a child on a swing. A slight push, just enough to overcome the friction forces, keeps the child swinging.

In a simple system consisting of a mass and a restoring spring, like that shown in Fig. 1, the mass oscillates in simple harmonic motion at its natural frequency when deflected from its neutral position and released. With friction in the system, the amplitude diminishes and the oscillation is slowly damped. But by employing an impressed torque, to compensate for the energy absorbed in friction, the oscillation can be sustained. The amplitude can be altered by changing the magnitude and duration of the impressed torque. The system is free in amplitude, but fixed in frequency.

In a system similar to that shown in Fig. 1 where

θ = angular displacement from the neutral position, radians
θ' = first derivative of θ with respect to time
θ'' = second derivative of θ with respect to time
$I\theta''$ = inertia acceleration factor
A = coulomb damping factor
$B\theta'$ = viscous damping factor
$C(\theta')^2$ = windage factor, varies as the square of the angular velocity
$K\theta$ = spring response factor

when undergoing forced vibration, the impressed torque is

$$I\theta'' + A + B\theta' + C(\theta')^2 + K\theta$$

At resonance, the inertia acceleration factor $I\theta''$ is equal and opposite to the spring factor $K\theta$ so that they cancel each other. To maintain oscillations the impressed torque must compensate only for the power dissipated by coulomb, viscous, and windage damping, and spring and bearing friction.

The impressed force required to overcome the dissipating forces should preferably be

in phase with the natural oscillations. If the impressed frequency varies in phase relationship, the ideal efficiency of the system is lost.

When reducing this theory to a practical application, the inertia mass must be free about its axis of oscillation, restrained by springs of proper restoring characteristics, and free to be "kicked" at the proper times with just enough energy to sustain oscillation at the desired amplitude. The spring must merely provide a restoring torque when deflected from neutral. For simple harmonic motion the restoring torque must be linearly proportional to the angle of deflection. The ration of restoring torque to angular deflection must be of the proper magnitude to provide the desired frequency with the associated inertia mass. Where

ω_n = natural circular frequency, oscillations per 2π sec.
f = oscillations per sec.
K = torsional spring constant, torque to produce an angle of twist equal to one radian
I = moment of inertia of the oscillating mass

$$\omega_n = 2\pi f = \sqrt{K/I}$$

In machines or apparatus where millions of oscillating cycles are performed in relatively short time intervals, the springs must be carefully designed to avoid mechanical failure by fatigue.

The principles discussed in the previous paragraphs were incorporated in the design of the MRS (mechanical resonant scanner) that resulted from an investigation of rapid scanning techniques undertaken at Radiation Laboratory, based on work done for the office of Scientific Research and Development under contract No. OEMsr—262. In airborne radar systems, the faster the airplane is travelling, the more rapidly do the ranges of the indicated targets change. The beam, therefore, must scan rapidly to supply "up-to-date" information on range and bearing of targets. The object of the investigation was to develop a scanner capable of 12 or more scans a second. At this rate, the image would appear as a moving picture rather than a series of unconnected pictures.

The mechanical resonance scanner is a system vibrating resonantly in torsional oscillation. The impressed frequency follows the natural frequency. The phase of the impressed force is determined from the oscillation rather than from an external source, therefore, the system is self-synchronous. When the natural frequency drifts, so does the impressed frequency, and the ideal performance persists.

In the mechanical mock-up of the MRS, Fig. 2, a wooden board approximates the windage of the radar antenna, and a steel disk has a moment of inertia equivalent to that of the antenna. Springs having the proper torsional constant provide a natural frequency about the axis equal to that desired as the scanning frequency.

Two tension springs were used in the initial tests and prototypes. The displacement curve of a single tension spring is shown in Fig. 3(A). The arrangement of the two springs as used is shown in Fig. 3(B), and also the linear torque factor resulting from the combined torque of both springs.

The "kick" is provided by a motor geared to the oscillating system. In this arrangement the motor armature is part of the oscillating inertia mass and must be considered in frequency and spring calculations. When referred to the "one speed shaft" the armature inertia is multiplied by the square of the gear ratio.

With a given motor, there is an optimum gear ratio for each scan rate and for each sector angle. In the MRS a gear ratio was chosen that was a compromise to give good results for the most used scan angles, since the scan frequently was fixed.

The motor on period is centered about the midpoint ("at rest" position) of the sector as shown in Fig. 4. When the motor is off the scanner coasts. The springs reverse the motion at the end of the swing. During the reverse swing the motor is turned on again in the opposite direction. Since the motor is revolving almost at rated speed when the current is turned on, there is no surge in the starting current. The oscillograms in Fig. 5 indicate the motor current duty cycle. Also, there is no surge in the reversing relay, since the polarity is reversed at the end of the swing, when the motor is off.

Control of the motor revolves itself into two problems: (1) Control of polarity so that the motor turns in the correct direction, and (2) control of the on-off cycle so that the "kick" is of the proper duration. Each of these problems has both a mechanical and an electronic solution.

The mechanical polarity control is simply a contact arm mounted loosely on an oscillating shaft. As shown in Fig. 6, movement of the contact arm is limited by two stops, one of which is an electrical contact. Friction between the arm and the shaft forces and holds the arm against one or the other of the

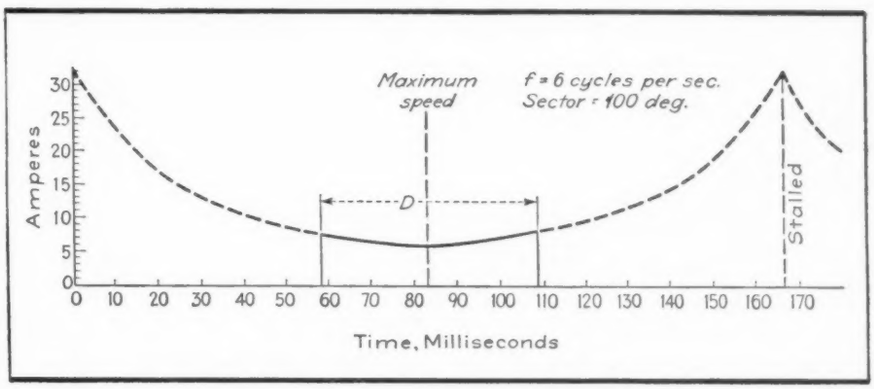


Fig. 4—Characteristic curve for motor current in the mechanical resonance scanner system. Motor is series wound. Interval D indicates time of current flow.

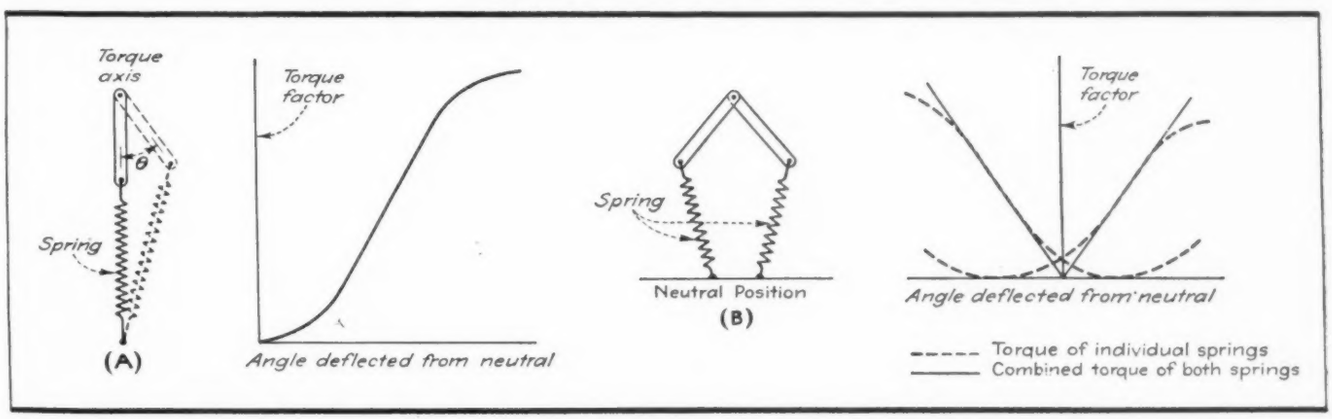


Fig. 3 (A)—Single tension spring and torque factor-deflection curve. (B) Two spring arrangement and resutlant linear relation between torque factor and angular deflection.

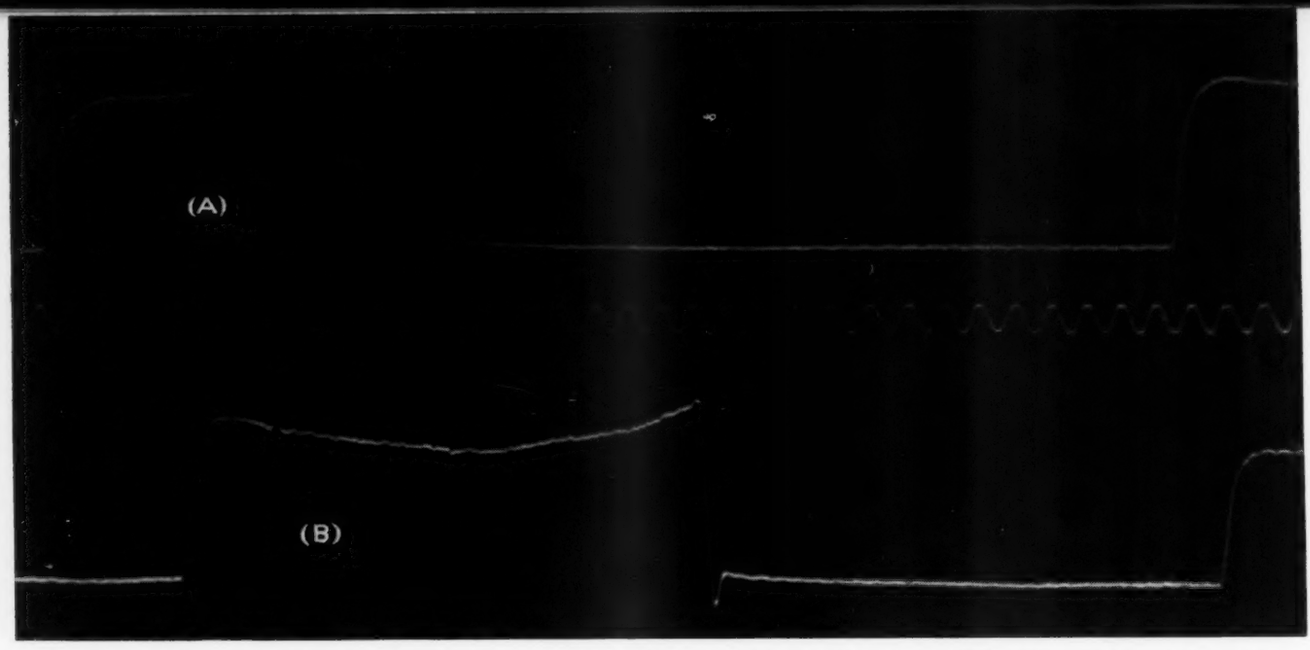


Fig. 5—Oscillograms of motor current duty cycle. (A) 70 deg. sector at 28 volts. (B) 100 deg. sector at 28 volts.

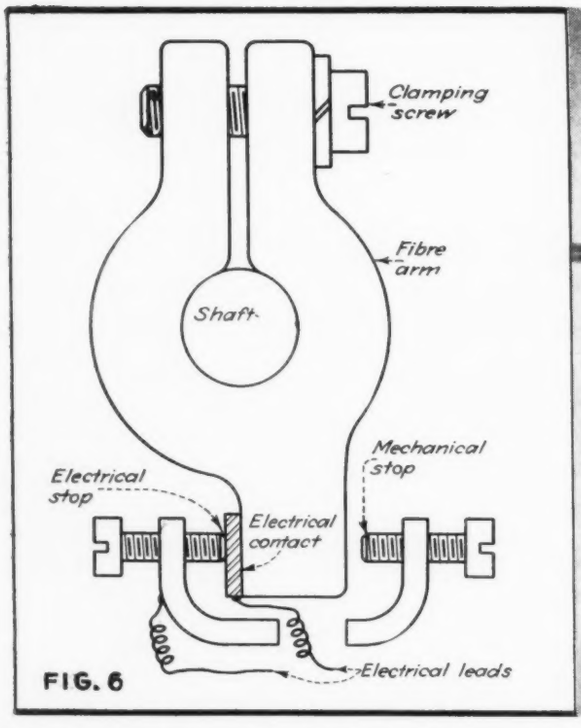


FIG. 6

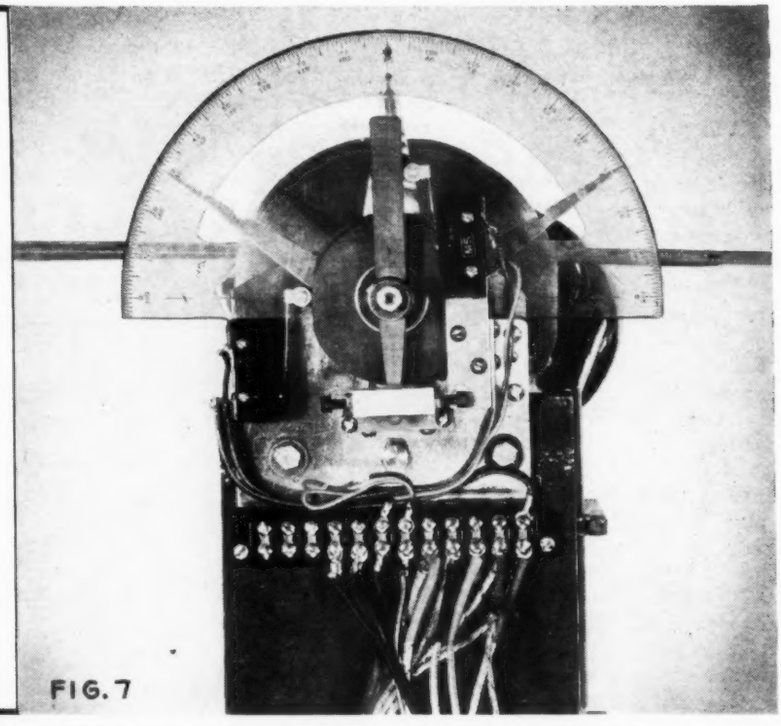


FIG. 7

Fig. 6—Contact arm of mechanical polarity control is mounted loosely on shaft with just enough friction to hold arm against either stop. Fig. 7—Front end of mock-up showing mechanical polarity control switch and sector angle control switch.

stops, depending on the direction of rotation of the shaft.

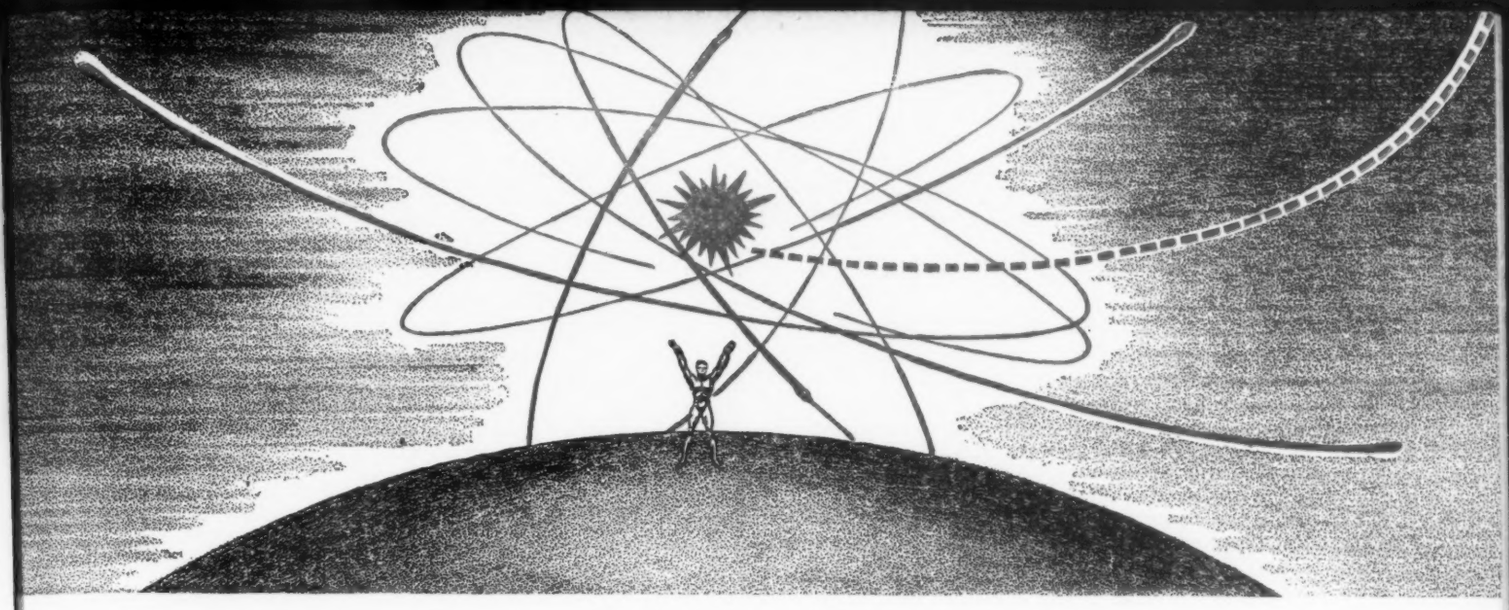
When the arm is against the electrical contact a circuit is closed that includes the coil of a reversing relay, thus selecting the proper polarity. A time delay is introduced at reversals, because the shaft must change direction before the relay closes and the polarity reversal takes place. Since the delay occurs while the motor is off, this method of switching offers no difficulty.

If it were desired to scan more than sixteen times a second, the delay time in the mechanical control could become an appreciable part of the cycle. To anticipate this difficulty, an electronic polarity control was devised using a phase shifting circuit that anticipates the reversal and thus balances the inherent delay of the reversing relay. With the electronic control, armature reversals can be made to occur exactly at the end of the swing, therefore, the velocity of switching would not be a limiting factor in increasing the scanning frequency.

The electronic control weighs more than a pound, its mechanical counterpart weighs two ounces. Thus for airborne use, where weight is a consideration, the mechanical arrangement should be used wherever permitted.

A series switch actuated by a cam on the oscillating shaft controls the on-off cycle of the motor. A different cam is needed for each desired sector angle. An electronic sector control is used on the prototype, and although heavier than the mechanical sector control, it provides a continuously variable sector from 10 to 155 deg. Merely turning the knob of a potentiometer controls the sector angle, this can be done with the scanner in operation. The view of the mock-up shown in Fig. 7 shows both the mechanical polarity control and the mechanical sector control arrangements.

[Author's Note: The author acknowledges with appreciation the interest and cooperation of Dr. Willoughby M. Cady, Rudolph Sher, Ensign Clarence W. Schultz, David L. Brewer, and Ralph I. Robbins in the development of this system.]



MAN vs ATOM—YEAR 1

WHEREIN WE SIGNALIZE THE FIRST ANNIVERSARY OF THE ATOMIC AGE, CONSIDER THE ALTERNATIVES INHERENT IN BOTH GOOD AND EVIL POTENTIALITIES OF NUCLEAR FISSION, THEN VENTURE A GLIMPSE INTO THE FUTURE

A YEAR AGO, July 16, 1945, at Alamogordo, New Mexico, man created the first atomic explosion. Most impressive events diminish in stature as they recede in time. This one grows bigger with each passing day. It truly marked the beginning of a new age.

As Year 1 of the Atomic Age ends and Year 2 begins, we are engaged in three portentous projects.

At Bikini Atoll we are detonating the fourth and, possibly, the fifth atomic explosions in the history of the world.

At Oak Ridge, Tennessee, we are building the first atomic energy plant for peaceful purposes.

Most important, in New York we and all the other United Nations are engaged in the first attempt to subject atomic energy to international control. Literally, the fate of the world hangs on this attempt.

As this introduction is written, the United Nations Atomic Energy Commission has just begun its work. People everywhere pray for its success—for their own sake, but even more for their children and for their grandchildren. If this Commission fails let everyone everywhere be warned: the world has taken a step toward destruction.

As we enter the second year of the Atomic Age, the nations of the earth are embarked on an atomic armaments race. There is no blinking that fact. We have had official notice served on us. Therefore, we must understand that unless the United Nations Commission can arrest the drift of events, we are moving toward a horrible war. The Commission must succeed.

The American delegate, Mr. Baruch, has brought to the Commission an ably thought out plan. It would internationalize nuclear science, and release for mankind the beneficent applications of atomic energy. But it would "control" atomic bombs only to the extent of giving the world brief warning of any nation's preparation to use them, so that we might have foreknowledge of disaster.

Therefore, the real and enormous task before the world becomes clear. We must end war. No other control of atomic weapons exists. If war comes, atomic weapons will be used. If they are used, our children who survive will curse their fathers. Understanding the consequences of failure, we *must* succeed.

Because we cannot succeed without knowledge, I have asked my associates at McGraw-Hill to condense into the following pages what we know at the close of Year 1 about this great new atomic force — its basic science, its possible uses and its political repercussions.

James H. McGraw, Jr.

President, McGraw-Hill Publishing Co., Inc.

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This Fateful Atom...

LOOKING BACK twelve months to the birth of Year 1, Atomic Age, we begin to sense the majestic import of the atomic bomb that blasted the naked desert at Alamogordo, N. M., on July 16, 1945. There man first shattered atoms in an explosive *fast-chain* reaction. Then came Hiroshima and Nagasaki.

In every case the fateful atom was either uranium 235 (U235), or plutonium derived from the action of U235 on U238. Every pound of U235 atoms split in these unprecedented blasts yielded the energy of 11.4 million kilowatt-hours, or 1400 tons of coal — slightly more for plutonium.

No matter where one mines uranium ore, the purified natural uranium (Fig. 1) always contains 99.3% of the "garden" variety U238, and a mere 0.7% of the precious U235.

An atom is like our solar system. The central sun is the nucleus — a bunched mass of protons and neutrons, each weighing one unit. The planets are electrons. Each proton has one plus electrical charge — each electron an equal negative charge. There must be as many negative electron planets as positive protons in the nucleus. This is also the "number" of the atom. Neutrons have no charge, but add weight.

The atomic number of uranium is 92 because the uranium atom always has 92 nuclear protons and 92 electron planets. The isotopes U238 and U235 differ only in the number of neutrons; U238 has 146 neutrons, and weighs 92 + 146 = 238 units. U235 has 143 neutrons, and weighs 92 + 143 = 235 units.

Ordinary chemical reactions, such as TNT explosions, release only a fraction of the modest energy of the whirling electrons in the outer atom. Nuclear reactions unlock the immensely greater energies which bind together the nucleus.

Even the gentle tap of a slow-moving neutron bullet will split the atom of U235 or of man-made plutonium into two medium-weight atoms, yielding also one to three spare neutrons plus energy. Thus these *fissionable* materials supply both their own bullets and a highly sensitive lot of high-explosive targets — a perfect setup for a *chain reaction* (Fig. 2).

Chain reactions work like chain letters. Neutrons from one nuclear explosion hit and explode other nuclei. But, since atoms are mostly open spaces a chain started in a small block of U235 or plutonium quickly dies out because most of the released neutrons escape from the block.

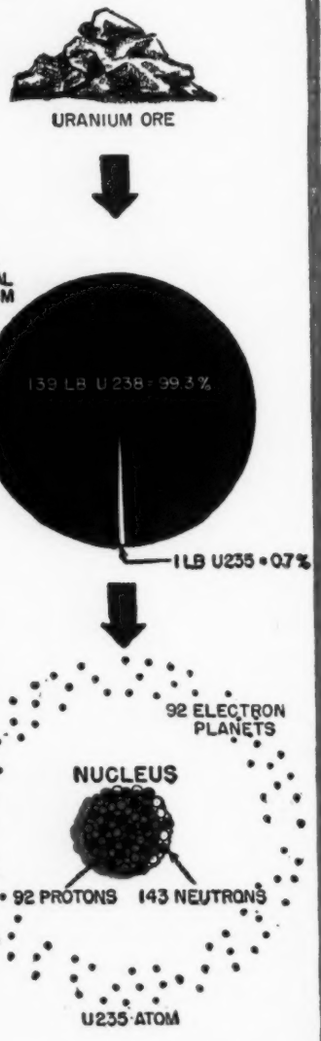
The bigger the block, the smaller will be the percentage of escaping neutrons, and the more left to split other nuclei. When the block is rapidly built up beyond a certain secret size the fragments of 1000 nuclear fissions split many more than 1000 additional nuclei. Then fissions multiply geometrically, and the block disintegrates with explosive speed and violence — as in a bomb (Fig. 3).

This bomb explosion is a fast-neutron chain. For economy and ease of control, uranium piles for the gradual release of nuclear energy for commercial purposes will normally use a lean fuel — that is U235 or plutonium diluted with U238, thorium or other less costly materials.

To maintain a chain reaction such piles must be large and artificially stimulated by using carbon blocks or some other *moderator* (Fig. 4) to slow many of the neutrons. Slow neutrons make more hits than fast neutrons because there is more time for them to be swerved from a straight path by the attraction of nearby nuclei, as shown below.

1 ORE TO U235

Only 0.7% of natural uranium is U235

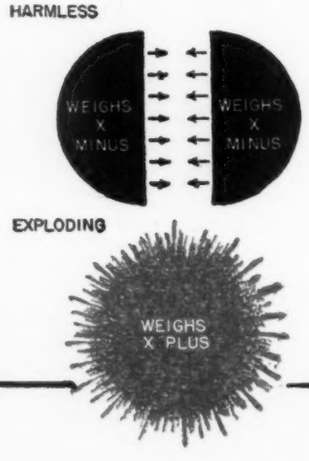


2 CHAIN REACTION

Fragments from earlier nuclear explosions smash other nuclei

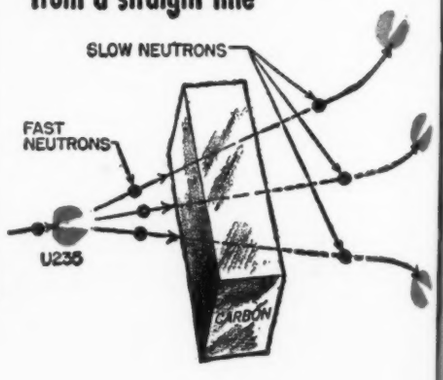
3 WHY BOMB EXPLODES

When block of rapidly assembled U235 passes secret critical size it explodes spontaneously



4 SLOW NEUTRONS MAKE MORE HITS

A slow neutron is more easily swerved from a straight line



can Serve Man . . .

THE FATEFUL U235 ATOM can serve man as a new, compact source of heat energy for power generation, comfort heating or industrial processing. Peacetime applications of atomic energy will use dilute U235 or plutonium as a "fuel," mixed with carbon or some other moderator to slow some of the neutrons and thus keep the chain reaction going.

The diluting agent may be either U238 or thorium, or both. These will do double duty, because neutron bullets convert U238 into the energy-yielding plutonium, and thorium into U233, which may prove equally serviceable.

Thus the commercial piles of the future will "burn" U235 to make other atomic fuels, plutonium and possibly U233, which in turn will deliver heat energy to the pile. In that way it will be possible to get from the pile far more heat than the equivalent of 1400 tons of coal for each pound of U235 split. This highly attractive prospect will speed the day when nuclear energy can compete with coal.

While already mechanically obsolete, the piles making plutonium for bombs at Hanford, Wash. (Fig. 1) reveal the basic principle on which future piles for power and heat will operate. The heat now wasted in vast quantities will be put to work. The plutonium, now removed for bomb manufacture, will be returned to the pile (or left in) as supplementary fuel.

ATOMIC POWER

The possible everyday applications of nuclear heat pictured in Fig. 2 have been recognized from the very first day of the Atomic Age. Year 2 will see the building of the world's first atomic power plant (a pilot plant) at Oak Ridge, Tenn.

Beyond question such installations will produce power, but it may be years or decades before they prove economical. To compete with conventional plants the piles must first be redesigned to run at temperatures high enough for good power-plant efficiency. Also the techniques of operating piles by remote control through the heavy radiation screens must be radically streamlined.

The Hanford piles run on natural uranium containing only 0.7% of U235. The typical commercial atomic power plant of the future will use more than 0.7% of U235 or plutonium, but less than 50%. This will avoid both the low efficiency of the too-lean mixture and the excessive fuel cost of the rich mixture. It will permit piles of moderate size and take maximum advantage of U238 and thorium as potential sources of plutonium and U233.

One should not expect U235 to replace coal generally in this generation, although a few central power stations and ships will try it out before Year 10 of the Atomic Age. Plants far from traditional sources of fuels may turn much sooner to uranium and thorium as concentrated heat sources, that may easily be transported even to remote corners of the earth.

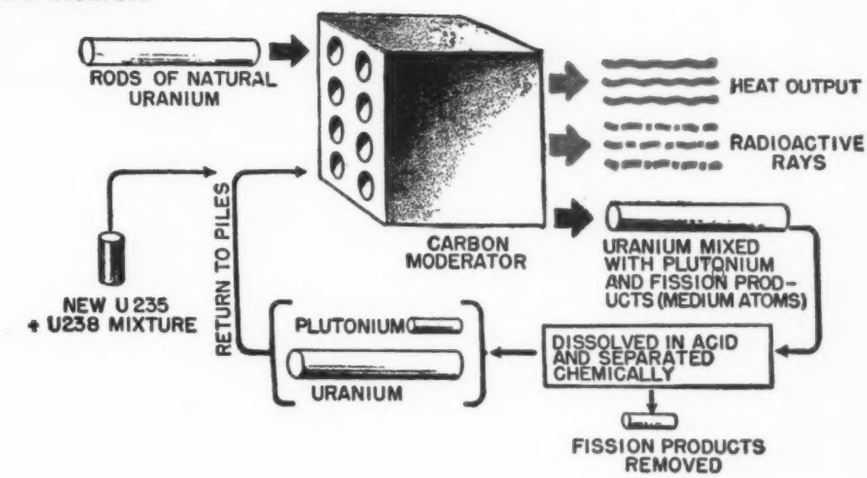
Atomic power, in forms now known, is impracticable for automobiles and small airplanes, because of the large initial investment in uranium and the need to carry 50 tons of shielding to protect riders and pedestrians against the deadly radioactivity accompanying nuclear fission.

RADIOACTIVE ISOTOPES

More immediately important than the heat and power applications of nuclear energy are the services that the radioactive byproducts of pile operation can render. Because these materials act chemically like their ordinary non-radioactive cousins, but can be followed and detected easily, they are expected to play tremendously vital parts in medicine and biology. For more details, see the last page of this section.

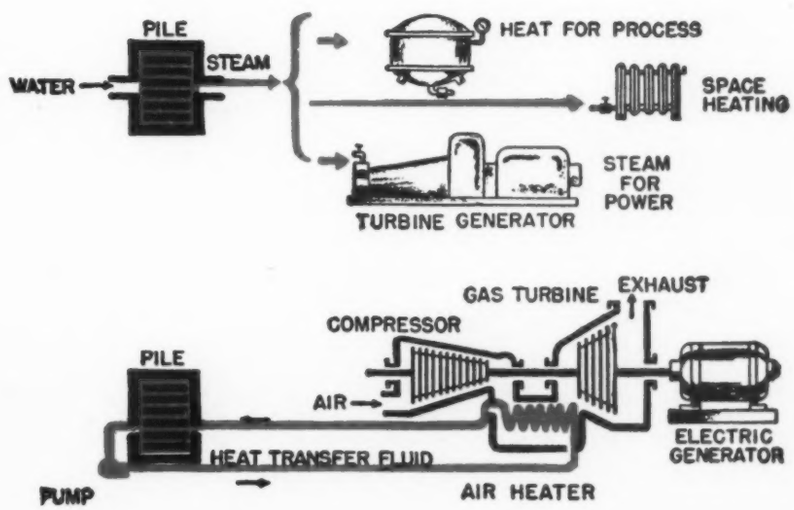
1 SLOW-NEUTRON PILE

Can make plutonium for bombs—or heat for power, process and comfort

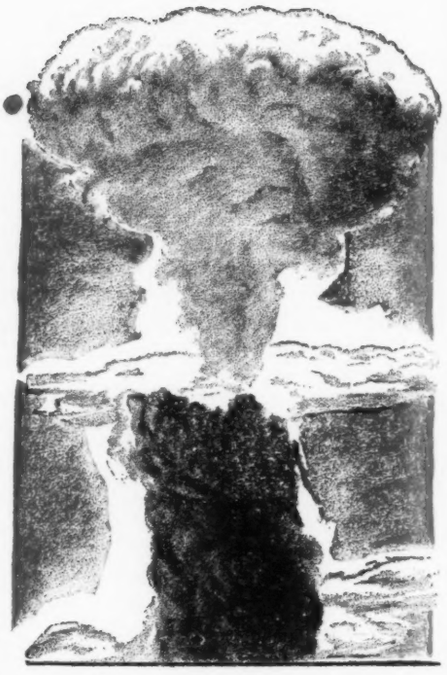


2 PRACTICAL APPLICATIONS

Include steam for turbines, process and comfort heating—also heat for gas turbines



or Destroy him . . .



A single improved atomic bomb can devastate ten square miles of city

URANIUM 235 and plutonium are now man's slaves. They will build or destroy as he orders. Man dreads this vast force only because he distrusts himself. War is proof that man in the mass has never achieved self-control. He has always sought better weapons; yet the perfect weapon now brings him no satisfaction for he sees in the atom bomb his own destruction as well as that of his enemy.

The ultimate benefits of nuclear energy may well surpass its present terrors, but the terrors are here now in awful dimension, and man must face them. He must pay this price for unlocking the wealth of the inner atom.

ATOMIC BOMB

This page, then, is about the atomic bomb. Nothing will be said here that is not either a certified scientific fact or a conclusion shared by the majority of the leading scientists, engineers and statesmen who have studied the matter.

As already explained, an explosive nuclear chain reaction spontaneously sweeps through a block of U235 or plutonium when the block is rapidly enlarged beyond a certain "critical" weight X. That weight is still a military secret; the official Smyth report vaguely suggests that it is more than 4 lb and less than 200 lb. Each piece of U235 in the dormant bomb must weigh less than X. At the desired instant of explosion the bomb mechanism assembles these pieces rapidly into a single piece considerably heavier than X.

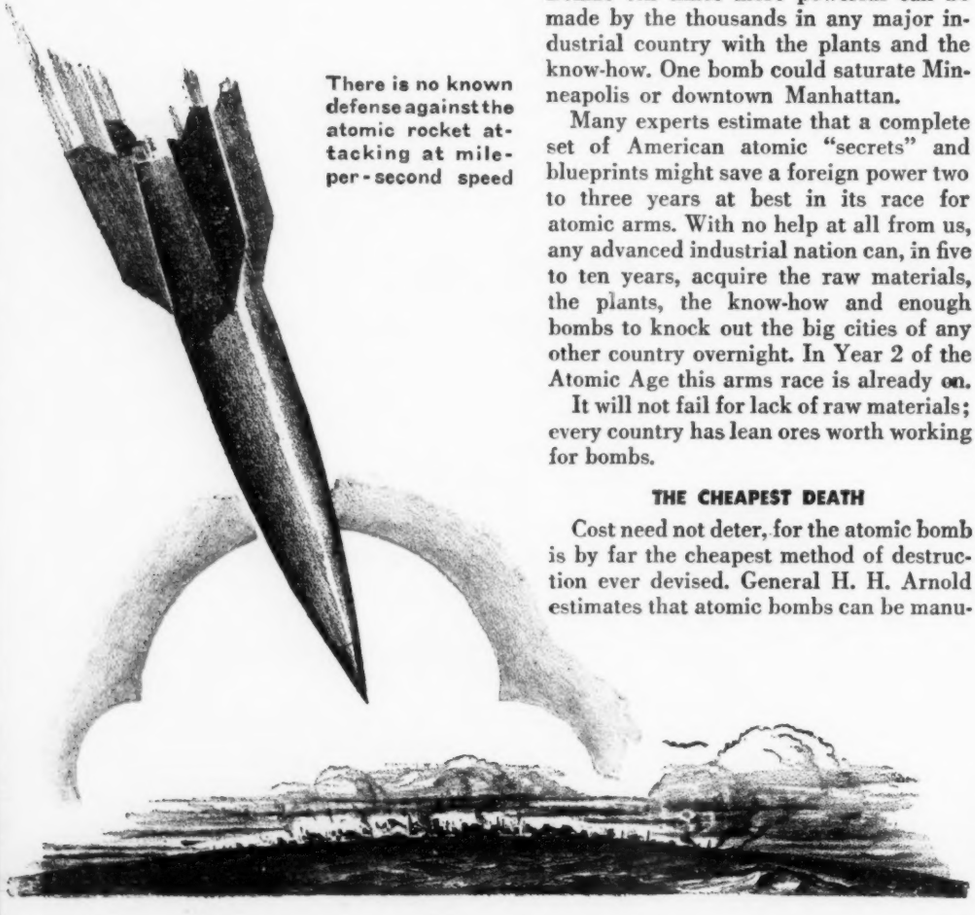
The explosion itself drives the U235 pieces apart, thereby quenching the atomic conflagration before all the atoms are split, so the bomb efficiency is far less than 100%. For each pound of U235 (or plutonium) atoms actually split, the bomb releases the energy of 1400 tons of coal.

This explosion is mainly ordinary heat at work in unprecedented concentration. Bomb metals become incandescent vapor millions of degrees hot. This, and the enveloping sphere of glowing air, radiate a blinding flash that chars human flesh at half a mile and blisters at over a mile. There is a destructive shock wave (sound) and a second-long hurricane of unimaginable force — the outrushing of the expanding heated air. Deadly neutrons and gamma rays speed out from the bomb.

A single atomic bomb killed about 100,000 at Hiroshima. Fewer died at Nagasaki only because the circle of potential destruction included much vacant land. Bombs ten times more powerful can be made by the thousands in any major industrial country with the plants and the know-how. One bomb could saturate Minneapolis or downtown Manhattan.

Many experts estimate that a complete set of American atomic "secrets" and blueprints might save a foreign power two to three years at best in its race for atomic arms. With no help at all from us, any advanced industrial nation can, in five to ten years, acquire the raw materials, the plants, the know-how and enough bombs to knock out the big cities of any other country overnight. In Year 2 of the Atomic Age this arms race is already on.

It will not fail for lack of raw materials; every country has lean ores worth working for bombs.



There is no known defense against the atomic rocket attacking at mile-per-second speed

THE CHEAPEST DEATH

Cost need not deter, for the atomic bomb is by far the cheapest method of destruction ever devised. General H. H. Arnold estimates that atomic bombs can be manufactured *and delivered* for less than $500,000 per square mile of destruction.

Don't be misled by the two billion dollars America spent on a project that dropped only two bombs on the enemy. New plants can be built at a fraction of wartime cost, and the investment spread over thousands of bombs, not just two.

NO DEFENSE

So the bombs can be made in ample quantity and paid for, but can they be delivered? The answer is: "Yes; by the time the bombs are ready they can be delivered anywhere and overnight." If the defenses of the target country are weak, piloted planes can get through in ample number. Ten percent would be enough.

For more effective delivery radio-steered pilotless planes and rockets can carry the atom bombs faster than sound. Such weapons will be almost untouchable by either antiaircraft artillery or manflown fighters.

Greatest threat of all will be the transoceanic rockets. The German V-2 rocket, which never once was stopped by Britain's defenders, points one way. It needs only transatlantic range (with atomic propulsion) and an atomic bomb in the nose. Forty-six feet long, loaded with 7500 lb of alcohol fuel and 11,000 lb of liquid oxygen, the V-2 of World War II rose 60 miles in the air and arced 200 miles in five minutes to deposit one ton of TNT in London.

Seeing so many strange things come to pass, the man in the street cannot distinguish between possible miracles and the impossible variety. From the very start of the Atomic Age he has been hoping for a "ray" that will explode the atom bomb far off. Competent scientists and engineers say that cannot be.

The only way to bring down a 3500-mile-per-hour rocket at a safe distance is to chase it with your own 4000-mph rocket. You can't win at this game often enough to establish ironclad protection.

The only specific defense against the atomic rocket known in Year 2 of the Atomic Age is to disperse all cities and put key industries underground. This would be very costly in time, money and national morale.

MORE AND BETTER BOMBS?

Some will ask whether the U.S., as the most powerful industrial nation, could not build more and better bombs and carriers than any other nation. Probably yes, but there is still no real security. If the "weak" opponent has enough atomic weapons to destroy us once, what advantage is there in being able to destroy him twice?

Shooting first could protect us now, but not after the world is atomically armed. If we were to destroy the enemy's cities, we would probably miss his well-concealed and protected bomb magazines and rocket launchers. A few minutes later he could return the atomic fire. In brutal simplicity, that is the picture of future atomic war. Everybody loses.

At this point one grasps at another straw: "If everybody is to lose who would be so foolish as to start an atomic war? And didn't the Germans refrain from using gas for a similar reason?" Possibly yes. It *may* work that way. But in a world atomically armed to the teeth some nervous finger *may* pull the fatal trigger.

ONLY ONE WAY OUT

Throughout history each new offensive weapon has called out its appropriate defense. But now the offense leaps centuries ahead in a single bound and the defense lies almost helpless everywhere, unless some technical protection, unknown as Year 2 begins, can be devised.

The situation is extremely dangerous. There is no clear way out except through some sort of international action first to stop the atomic arms race and, before it is too late, to hobble war itself.

Can it be done? Perhaps not, but there is no alternative except atomic chaos.

...so he faces the Atomic Dilemma...

THE NUMBERED statements that follow in somewhat logical pattern are too fateful to be accepted on anybody's say-so. Every reader should test them in the light of his own information and understanding.

The points below sum up the conclusions of the previous article — and these in turn reflect a great mass of thought and discussion among leading scientists, engineers and statesmen close to the problem. To an amazing degree they concur on both facts and conclusions. For authoritative statements of their line of thought, in detail not possible here, the reader should see the recent book, *One World or None.*

THE DILEMMA

Nations must either face the probability of an atomic World War III, which would surely be the most deadly in history . . .

***Or,* the experts propose, yield both atomic weapons and war potential to international authority backed by superior force.**

What the Experts Say

1. In five to ten years any major industrial nation can make enough atom bombs to destroy all the major cities of any other country overnight.

2. This assumes no "secret" information or other help from us.

3. The necessary uranium ores will be at hand.

4. The cost will not be too high.

5. The bombs produced can then be carried thousands of miles by bombers, or by atomically powered guided missiles moving faster than sound.

6. There will probably be no effective military defense against such weapons.

7. Dispersing cities, and putting key industries deep underground, will give some protection if accomplished in time, but at incredible cost in money and human discomfort.

8. In a world atomically armed, nations can probably protect their bomb stocks and rocket launchers from enemy assault.

9. If so, nation A can destroy the cities of any other nation B, after which B's rockets will destroy the cities of A. Shooting first will not win an atomic war.

10. This knowledge may not restrain the trigger finger of a suspicious power.

11. Having more and better atomic weapons than the other fellow won't help much if he has enough to destroy us. No use to kill a man twice or rebomb urban ruins.

12. Every nation is vulnerable in the Atomic Age, including the U. S. A.

13. National security will be impossible without (first) international control of atomic arms and (not too long thereafter) international control of all war potential, both backed by superior physical power.

14. If action to this end is long delayed, it may become impossible to halt the atomic arms race already started.

15. At best, the necessary degree of international control, with some real delegation of national sovereignty, will be a revolution in human affairs. It may prove to be humanly unobtainable at this time. If so, men and women everywhere must face the probability of an atomic third world war—by far the most destructive in all history.

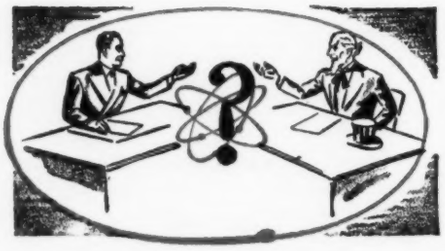
In this atomic age no nation can be safe through its own unaided might

..and the Great Debate unfolds

ATOM YEAR 1 has probably been marked by more debate on a single subject than any other twelve months in the world's history. Social, economic and political as well as purely technical issues have been pressing for realistic solution. Let us look at these issues and see where we stand:

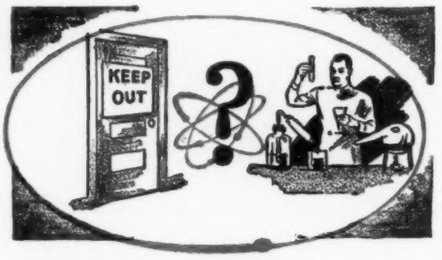
CIVILIAN VS. MILITARY

Because the atomic bomb is the world's greatest weapon, the armed forces would like to control it. But because atomic energy can also be used for peaceful, beneficial purposes, civilian control seems equally essential. These conflicting viewpoints had their strong proponents before the Congress which finally reached a fairly satisfactory compromise in the Atomic Energy Bill of 1946, setting up a competent civil board with which the armed forces will have continuing liaison. As we go to press, just before Year 2 of the Atomic Age begins, this bill has passed the Senate, but there is still a question how rapidly it will be enacted into law.



PRIVATE VS. PUBLIC

Atomic energy is "too big" and "too hot" to be handled privately. It must be nationalized and internationalized. The questions are *how* and *to what extent.* Fortunately, as the "boxes" on these pages show, there are means that may attain reasonable safety against misuse of the atom, and still do so without public control of many "non-dangerous" applications.



SECRECY VS. FREE SCIENCE

Throughout the first year of the Atomic Age hot debate has raged around "keeping the secret of the bomb." To prevent potential enemies from making atom bombs some have urged a complete blackout of all phases of atomic energy — even of the scientific fundamentals of nuclear physics. Others have sought immediate and complete disclosure of all bomb "secrets," both scientific and technological. These have held that such information cannot be effectively hidden, that secrecy blocks progress and breeds wars.

A year of debate has brought the great mass of vocal opinion to this middle ground: (1) Ease restrictions on the exchange of basic physical knowledge. (2) Release for industry's benefit many of the devices and methods developed for the bomb project. (3) Hold tight to specialized information on atomic bombs and bomb-material production until international safeguards are fully operative.

DOMESTIC CONTROL AS PLANNED IN THE ATOMIC ENERGY BILL OF 1946

McMahon Committee Bill contains the following provisions.

Policy. Declares it the policy of the U. S. to develop and utilize atomic energy to improve the public welfare, increase living standards, strengthen competitive enterprise and promote world peace.

Organization. Establishes the Atomic Energy Commission (AEC) of five administrators to direct four divisions on research, production, engineering, and military applications—to work in liaison with three committees from (1) the armed forces, (2) outstanding civilians, and (3) joint Congressional representatives.

Production. AEC to own and operate (under management contracts with industry if deemed desirable) all facilities for the production of fissionable materials, such products to be distributed with their radioactive byproducts under license for private industrial and medical research.

Military Application. AEC to engage in development work and produce atomic bombs as directed by the President, to be delivered only on his order to the Armed Forces.

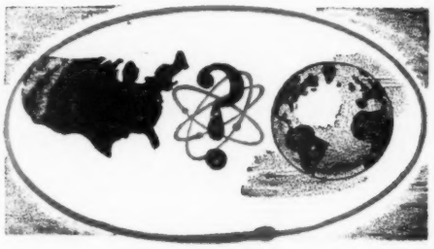
Industrial Utilization. Permits AEC to conduct research, design and manufacture equipment for atomic-energy utilization, license its use, produce and sell power obtained as a byproduct in the production of fissionable materials. Directs AEC to give widest safe scope to private initiative.

Control of Information. AEC to enforce a ban on the dissemination of restricted data that might be used to injure the U. S. or secure advantage to a foreign nation, yet to provide leeway for ultimately relaxing restrictions as future conditions warrant.

Patents and Inventions. No private patents permitted for production of fissionable materials or their utilization for military weapons, but AEC will justly compensate for such inventions, when made by private citizens. Patents for non-military applications may be purchased or condemned by the AEC only when public interest is affected.

Appropriations. "Such sums as may be necessary and appropriate to carry out the purposes and provisions of the act" *plus* unexpended funds of the Manhattan Engineer District.

while Time runs out



NATIONAL VS. INTERNATIONAL

Born of nationalism, the Atomic Age began when three nations discovered a weapon that today gives them the greatest military power on earth. The prime question is: Shall the atom remain the servant of its conqueror, nationalism?

During Year 1 of the Atomic Age the Truman-Atlee-King declaration, the masterly report of the State Department's atomic consultants, and the U.S. representative on the United Nations Atomic Energy Commission, have all called for international control of atomic energy. Year 2 will start with no such control. This failure to decide and act is in part a natural result of the extreme difficulty of the problem and the obvious dangers of unwise decisions. Nations everywhere face a triple dilemma in this Atomic Year 2: the dangers of nationalism, the dangers of internationalism, the supreme danger of not being able to make any decision in time to meet the atomic bomb threat.

INTERNATIONAL CONTROL AS PROPOSED BY THE U.S. TO U.N. ATOMIC COMMISSION

Baruch statement follows constructive path laid out by Atomic Consultants in "Acheson-Lilienthal Report."

The Plan. The U. S. has proposed that all nations band together to outlaw the use of atomic energy for war and to promote and harness its development for the benefit of mankind. To this end an International Atomic Development Authority would be set up, and to it the U. S. would turn over, at various stages of its organization, all atomic bombs, know-how, raw materials, facilities, and stockpiles of fissionable material. Thus IADA eventually would supersede national authorities on some matters and supplement them on others.

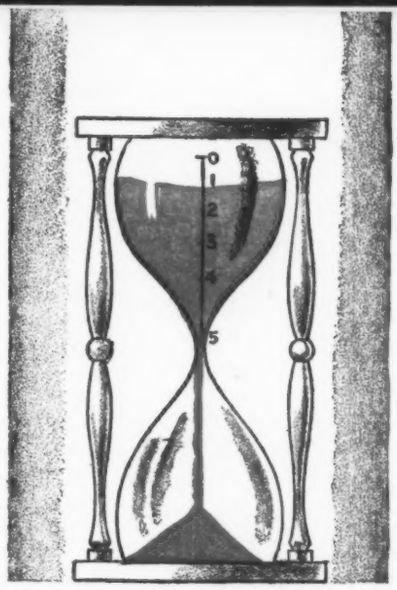
Owner and Operator. IADA would take over from national authorities or private ownership full management and control of all atomic energy matters that afford a possible threat to World security. These include:

1. Raw Materials—Supplies of uranium and thorium to be inventoried, controlled, and developed by IADA.
2. Facilities—IADA to control and operate plants producing fissionable materials and to own and control their products.
3. Research—IADA to undertake research and development on all aspects of atomic energy and to possess exclusive right of research on atomic explosives.

Private Initiative. Will have its chance to push forward the use of atomic energy for peacetime (non-dangerous) purposes. With IADA providing raw materials and carrying out necessary inspection, national and private enterprise may operate "safe" power piles, and produce and use radioactive isotopes for research, clinical and other applications. Radioactive isotopes produced by IADA also can be distributed for peacetime use.

The Mechanics of Safety. No plan is a certain guarantee against future atomic war. This plan should, however, prevent surprise attack with atomic weapons; for IADA is to buttress positive ownership or management controls with wide powers of inspection. Obviously, successful inspection rests on complete freedom of access or egress in any area.

Sanctions. At the heart of the plan lies the problem of penalty for violation — a matter for profound statecraft. To the U.S., one aspect of sanctions appears crystal clear: Here is an area where the veto right now held by the five great Powers must be redefined if it is not to be incompatible with the meaning and purpose of the proposed control.



Leading industrial nations can produce atomic bombs in five years, competent scientists announced after Hiroshima. Already one year of the precious five has been consumed in debate without international action. Soon it may be too late to check the growing momentum of the atomic arms race.

TIMETABLE—ATOM YEAR 1

1. **July 16, 1945. World's first atomic bomb detonated in New Mexico.**
2. **July 26, 1945. President Truman and Prime Minister Churchill issue Potsdam ultimatum threatening Japan's destruction if she continues.**
3. **August 6, 1945. Atomic bomb dropped on Hiroshima.**
4. **August 9, 1945. Atomic bomb hits Nagasaki.**
5. **August 11, 1945. Army releases Smyth Report on "Atomic Energy for Military Purposes."**
6. **August 14, 1945. Japan accepts terms of Potsdam declaration.**
7. **November 15, 1945. Truman-Atlee-King issue declaration of intention and procedures looking toward international control of atomic energy by United Nations.**
8. **March 28, 1946. State Department issues Acheson-Lilienthal Report on the "International Control of Atomic Energy."**
9. **April 12, 1946. Manhattan Engineer District announces program for experimental development of atomic power.**
10. **June 1, 1946. "Atomic Energy Bill of 1946" passes Senate unanimously, is referred to House of Representatives.**
11. **June 14, 1946. First meeting of United Nations Atomic Energy Commission (Bernard Baruch as American member). Manhattan District announces availability of radioactive isotopes for research use.**
12. **July 1946. Joint Army-Navy tests of atomic bombs at Bikini.**

. . . but if Man Masters Atom . .

IF MUTUAL DESTRUCTION by the atomic bomb can be avoided, the first century of the atomic age will bring immense advances in scientific knowledge, health and living standards. Already many prospective benefits can be outlined, but those we can neither foresee nor suspect may be even more important.

This prediction is grounded in scientific experience; the most fundamental discoveries have always been the most fruitful. The study of molecules gave us chemistry. Faraday's experiments with electricity and magnetism are the foundation stones of the great electrical industry. Can one expect any less from an understanding of the heart of every atom?

BENEFITS

Atom-splitting benefits clearly visible today fall mainly in three classes: (1) heat and power applications of the uranium piles; (2) general industrial applications of equipment and methods originally developed for the bomb project; (3) chemical, biological and medical uses of the "tagged atoms" (radioactive isotopes) now abundantly available from pile operation.

It is now evident that the energy yield of the U235 in an atomic pile can be multiplied many times by returning to (or leaving in) the pile the plutonium and possibly the U233 produced respectively from the U238 and the thorium in the pile. This is an indirect way to "burn" inexpensive U238 and thorium, and thus greatly extend the supply and reduce the cost of atomic fuels.

POWER APPLICATIONS

Although present piles run at low temperatures, it is certain that temperatures high enough for the efficient operation of steam and gas turbines will be attained. Already an experimental atomic power plant has been ordered. Atomic power for certain remote installations (say, for heating Arctic airports) may not be far off.

In five or ten years uranium piles will be driving a few experimental ships and submarines. In 20 or 30 years uranium may begin to compete widely with coal as a fuel for suitably situated large central heating and power plants. The 50-ton minimum weight of shielding rules out nuclear power for automobiles and small piloted planes.

SPECIAL USES

Some day ultra-high temperatures from splitting atoms will be used for special industrial operations on metals and other materials. Even the dread atomic bomb might easily serve peaceful ends — blasting lakes in deserts, changing the course of rivers, leveling mountains.

INDUSTRIAL BYPRODUCTS

The special industrial equipment and methods developed for the bomb project will find hundreds of important uses — mostly for purposes unrelated to atomic energy. These developments include pumps with neither seals nor leaks, leak detectors of amazing sensitivity, ultratight welding, a portable mass spectograph for quick and automatic gas analysis, new ways of handling corrosive and poisonous materials, new diffusion barriers for the separation of gases and of petroleum products.

TAGGED ATOMS

Yet more important than any of these, in the long run, will be the hundreds of radioactive isotopes now available as by-products of pile operation. Chemically indistinguishable from the ordinary forms of the elements, these isotopes serve as tagged atoms or "spies" if mixed with common stable atoms of the same species. They "fly with the flock," and can later be identified as surely as banded birds. With these amazing tools of research, the course of any element or compound may be traced through the bodies of men, animals and plants. Similarly, tagged atoms may be used in studying the course of many kinds of industrial and chemical operations.

BIOLOGY AND MEDICINE

A suspected hyperthyroid condition can be diagnosed by feeding the patient a minute measured amount of radioactive iodine. The click of a "Geiger" counter placed on the patient's neck will tell (1) what percentage of the swallowed iodine concentrates in the thyroid cells and (2) how rapidly that concentration is accomplished — giving a definite indication of the state of the gland.

In similar fashion the radioactive isotopes of hydrogen, oxygen and carbon will trace out the intricate transformations of carbohydrates and proteins in the human body. Radioactive phosphorus will explore the bones. Radioactive iron will show how and where blood cells are formed. Radioactive sodium will time the circulation of blood.

USES IN INDUSTRY

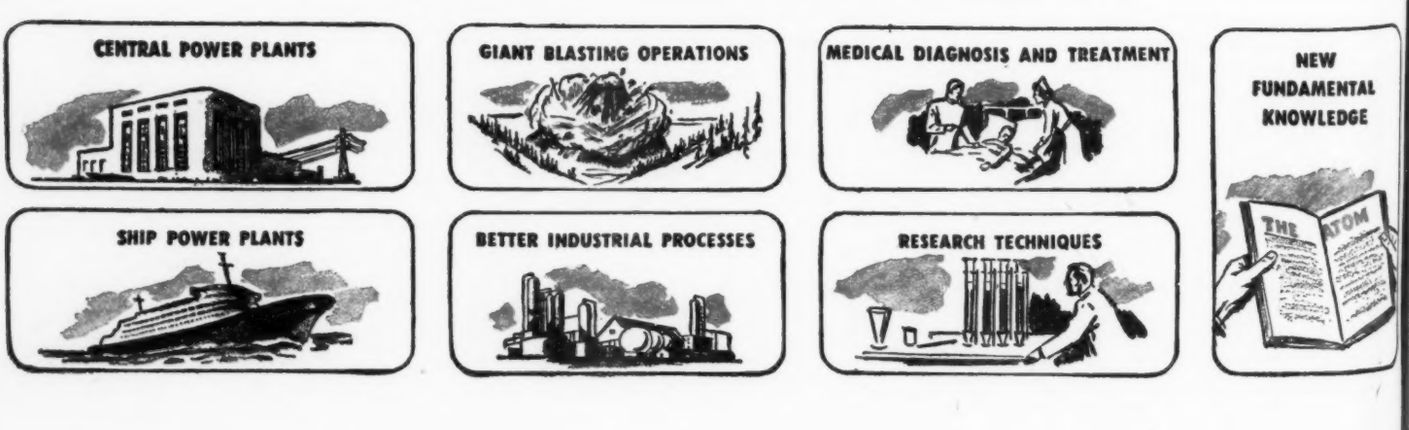
In chemistry the radioactive isotopes will speed the understanding of metallurgical and organic reactions. In industry they will measure flow, detect leaks, and do other useful work.

Meanwhile the uranium piles will be manufacturing certain radioactive isotopes that can serve as cheap but effective substitutes for high-cost medical radium.

KNOWLEDGE COMES FIRST

It is already clear that the chief benefits of atom splitting will come first as new scientific knowledge rather than as new engines and gadgets. But in the long run man's new understanding of the inner atom will enrich the whole range of human activity. This has always been the case with less fundamental discoveries in science. It can hardly be less with this most fundamental discovery.

ATOM SPLITTING WILL SERVE MAN IN:



ENGINEERING ABSTRACTS

FOREIGN ARTICLES—TECHNICAL SOCIETY PAPERS

Narrow Sleeve Bearings

Foreign Abstract condensed from "Sleeve Bearings as a Substitute for Ball and Roller Bearings," by Erich Fals in *VDI Zeitschrift*, October 14, 1944, page 573.

ALTHOUGH sleeve bearings of small width were suggested many years ago, their practical application has been a rather recent development. A sleeve bearing is considered narrow or short when the ratio of breadth b to diameter d is about 0.3 to 0.5. Two questions are most important in the development of short or narrow bearings:

1. Is it possible to design narrow sleeve bearings to have the same load carrying capacity as ball or roller bearings of the same nominal dimensions?

2. Is it possible to obtain satisfactory lubrication of narrow sleeve bearings?

LOAD CAPACITY OF NARROW SLEEVE BEARINGS. The most important disadvantage of all sleeve bearings lies in the impossibility of

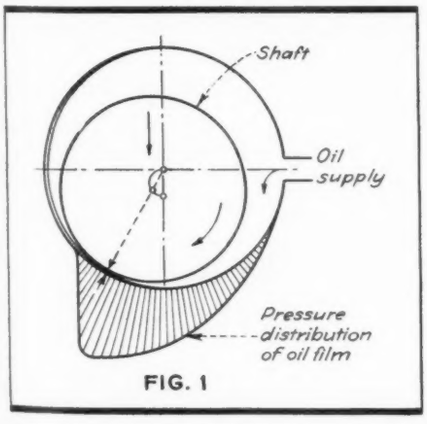


FIG. 1

Fig. 1—Pressure distribution in the oil film of a plain bearing.

obtaining uniform load distribution, as caused by the location of the oil hole.

It will be seen from Fig. 1 that, when rotating, the axis of a sleeve bearing is displaced by the non-uniformity of the pressure exerted by the oil film. The displacement varies with the applied load, the clearance between metallic parts, the oil viscosity, the relative laminar velocities within the oil film, and the boundary constraints of the metallic parts. Small sliding velocities are disadvantageous. The equilibrium of the forces results in a so-called "hydrostatic floating" of the journal of the wedge-shaped oil film and the load carrying capacity of sleeve bearings depends on the stability of the film.

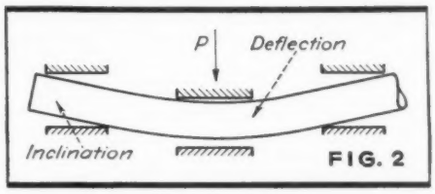


FIG. 2

Fig. 2—Illustration of "deflection" and "inclination" of the spindle.

To obtain narrow bearings with a load capacity comparable to that of ball or roller bearings, the pressure on the projected area of the journal must be relatively high, 1,500 to 4,500 lb. per sq. in. The "load zone" of the narrow bearing must, under such high pressures, be free from grooves, oil pockets and similar discontinuities of the surface.

Longitudinal grooves in the bearing reduce the load carrying capacity. Similar effects are caused by cavities, pores and cracks. But pores and light cracks that do not go through

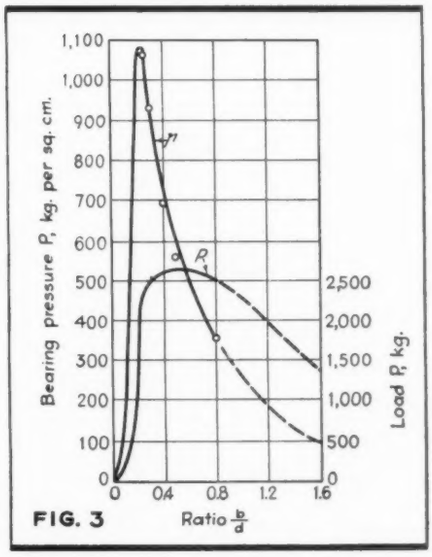


FIG. 3

Fig. 3—Effect of the b/d ratio on the maximum load capacity. Bearing material: Lead Bronze, Standard No. 25.

to the outside of the bearing are often desirable because they act as "storage room" for the lubricant.

Another consideration in the design of sleeve bearings is the deformation of spindles

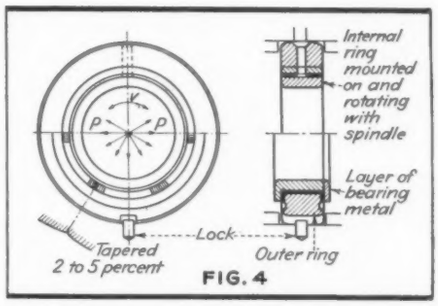


FIG. 4

Fig. 4—Double-ring SMT bearing for medium loads and high velocity; non self-lubricating type.

subjected to bending loads. The deformation exhibits itself in two ways; namely, the curved deflection at the center of the spindle, and the inclination at the spindle ends, as shown in Fig. 2. It is possible to compensate spindle angularity by use of self-aligning sleeve bearings. The only way to counteract the effects of spindle deflection within the bearing is through the use of narrow bearings.

The relationship between the b/d ratio of narrow bearings, unit bearing pressure, and the maximum load is shown in Fig. 3. The unit load carrying capacity of sleeve bearings decreases rapidly when the b/d ratio is less than 0.2 and likewise when it is more than 0.8. The most favorable b/d ratio lies between 0.3 and 0.8. All surface irregularities and roughness, of course, should be reduced as much as economically permissible.

LUBRICATION OF NARROW BEARINGS. Lubrication of narrow bearings is often more difficult than that of ball or roller bearings because of space limitations inherent in a narrow bearing. The lubrication method can only be determined when the direction of the load is known. There are two main types of lubrication:

1. Injection oiling by oilers, grease cups, oil pressure systems and other "non-self" lubrication methods.

2. Self lubrication by wick oiling, loose

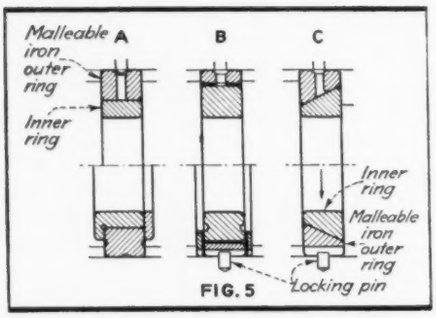


Fig. 5(A)—Double-ring SMT bearing made of sintered iron. (B) Similar bearing for high pressure and low velocity. (C) Single SMT bearing with tapered inner ring for slightly higher axial loads.

ring lubrication, and immersion lubrication.

Greasing is used when the bearings are operating in dust or water and particularly when the sliding velocity is low. For machine tools, centralized oiling systems and loose ring oiling for stationary machinery are recommended. Wick oiling can be used for small loads and velocities.

NON SELF-LUBRICATING NARROW BEARINGS. The two-ring narrow bearing shown in Fig. 4, is the "standard" design of a non-self lubricating narrow bearing. It consists of an

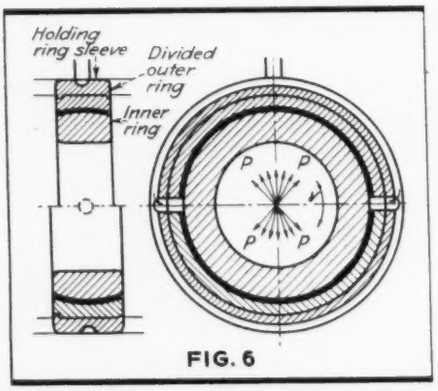


Fig. 6—Pendulum SMT bearing. Upper section: Standard No. 02, lower section Standard No. 22.

inner ring with a hardened surface and an outer ring that carries the bearing material. The top sectional view of Fig. 4 shows a transmission bearing, the bottom sectional view a guide bearing. The shoulders of the guide bearing project beyond the width of the outer ring, thus making an exchange with some of the ball or roller bearings series difficult. The reason for the shoulders is that for high-duty radial bearings the b/d ratio should not be less than 0.25. If the width were to be reduced to that of a ball or roller bearing, the b/d ratio would drop to 0.20. When substituting ball or roller bearings of standard series 22, 23 and 24 space conflicts do not arise, and the shoulders would not interfere.

The comparisons of bearing designs discussed in this article refer to anti-fraction bearings of series 02, that is a bearing with 40/80 mm. dia. and 18 mm. width.

Large axial loads cannot be taken by narrow sleeve bearings unless specially designed thrust rings are provided. Axial loads to 10 percent of the radial load can be taken by straight-faced rings, and those to 20 percent by tapered rings.

Three designs of two-ring narrow bearings used in machine tools are shown in Fig. 5, they are usually equipped with drip oiling. The bearing shown in Fig. 5(B) is particularly useful for low spindle r.p.m. and high bearing pressures. The I.D. is large and the wall of the outer ring is just as thin as rigidity permits. Fig. 5(C) is a special bearing for higher axial loads. It is a tapered narrow bearing with an outer ring of sintered iron. The load capacity is not known.

A swivel narrow bearing for loads in both directions, upward and downward, is shown in Fig. 6. The spherical surface of the inner ring runs in a correspondingly shaped outer ring that is split in the direction of the spindle and held together by a pressure ring. The oil enters at right angles to the two main load zones. This bearing is particularly suitable where oscillating bending loads cannot be avoided.

A universal narrow bearing for radial loads is shown in Fig. 7. Although the design does not completely correspond to the theoretical requirements, the bearing performs satisfactorily when the oil velocity is not too small. The oil is fed under pressure through three radial holes, not more than 0.08 in. dia., connected with a circular groove at the circumference of the outer ring. When high

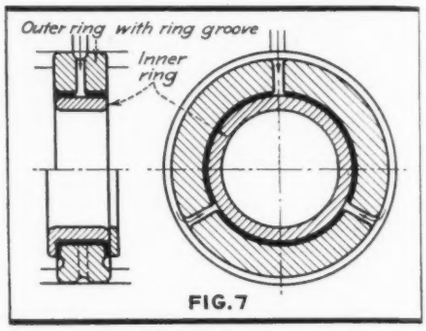


Fig. 7—Double-ring SMT bearing for radial loads in all directions.

spindle speeds are required, the heat dissipation can be increased by increasing the oil discharge section, that is, by increasing the clearance and by providing fine longitudinal grooves.

An interesting design that does not exceed the dimensions of ball or roller bearings is shown in Fig. 8. A thin, seamless covering made of plastic material is glued to the two tapered inner rings to facilitate heat dissipation through the outer ring. Such a bearing is superior to cylindrical bearings with plastic covering as it can be used for higher speeds, because of the more favorable sliding conditions. The heat resistance of the plastic material is particularly high.

NARROW BEARINGS WITH SELF-LUBRICATION. More difficulties must be overcome when designing self-lubricating narrow bearings than those previously discussed, because

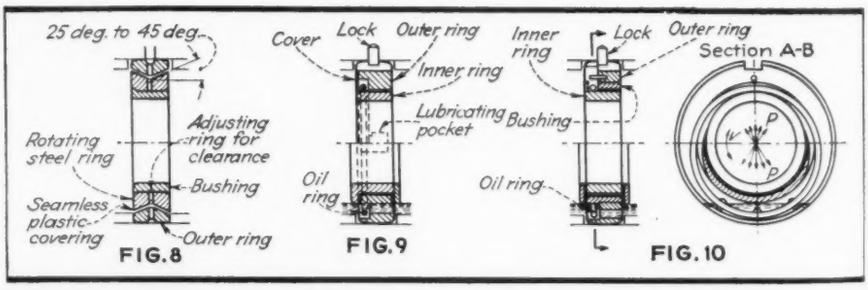


Fig. 8—Double-tapered SMT bearing with plastic covering for high axial loads. Fig. 9—SMT bearing with outer ring of special cast iron. Fig. 10—Bearing with pressed-in bushing.

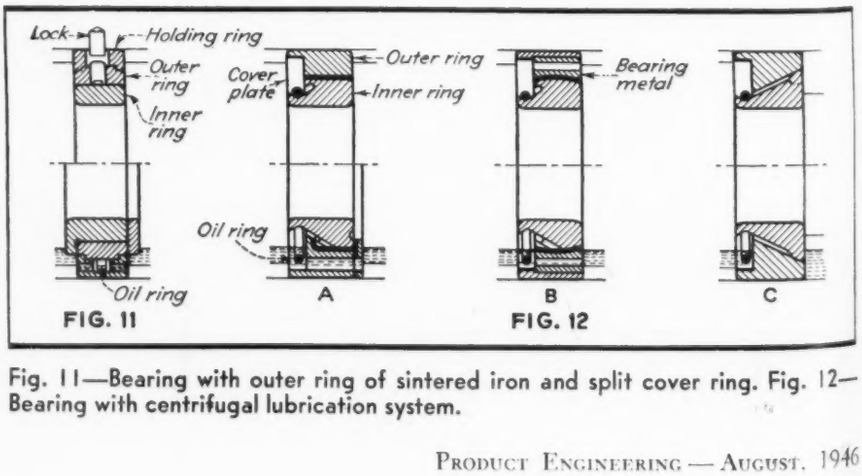


Fig. 11—Bearing with outer ring of sintered iron and split cover ring. Fig. 12—Bearing with centrifugal lubrication system.

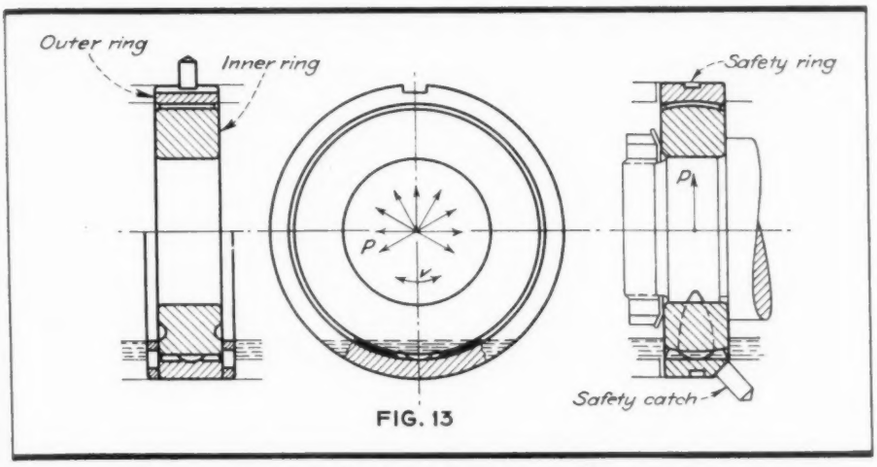


Fig. 13—Two-ring SMT bearing with immersion lubrication.

of the necessity of accommodating lubrication parts. Figs. 9 to 12 show narrow bearings with loose ring lubrication. Fig. 9 shows a design where the outer ring has a face groove for the oil ring while shallow oil pockets are provided on both sides of the center line. Fig. 10 is a similar design except that the oil ring rotates in a slot in the bushing. The bearing in Fig. 11 has a one-piece outer ring made of sintered iron and a two-piece holding ring that permits the assembly of the oil ring in the center of the bearing. In all these bearings, only small amounts of oil can be supplied because of lack of space and therefore appreciable slip occurs.

In the bearings shown in Fig. 12, the oil rings transmit oil to the tapered surfaces that lubricate the bearing surfaces. Fig. 12(B) is almost identical with Fig. 12(A) except for the spherical surfaces that permit self-alignment and also a swiveling of the inner ring for inspection of the bearings surfaces.

Immersion lubrication is used in narrrow bearings where the load is mainly acting in an upward direction as shown in Fig. 13. Wick lubrication was originally tried on narrow bearings without success. The humidity and decrease of oil viscosity reduced the efficiency of the wicks and this resulted in overheating of the bearings. A small wick oiled bearing is shown in Fig. 14. The wick groove in the outer ring is milled with a flat so that the inner ring can project with a slight crown. The wick drags on the inner ring under light tension that eliminates the possibility of injuring the wick. These bearings are dipped for a short time in warm mineral oil and then ready to use. They are simple, inexpensive and require little servicing.

An interesting type of narrow bearing, shown in Fig. 15 employs a vacuum generated in the oil film. The oil film sucks oil automatically when the bearing is "tapped" at a point beyond the area of minimum thickness of oil film. This reaction results from the fact that the width of the wedge-shaped oil film increases beyond the smallest section. Although this phenomenon was first investigated in Switzerland about 1919, its practical application is relatively new.

Tests with bearings of a b/d ratio of 1.0 proved the existence of a vacuum of 0.2 to 0.3 atmospheres. The narrow bearing is shown in Fig. 15(A) is designed for forces acting downward and for rotation in either direction. The lower portion of the bearing carries a wide oil groove from which two channels branch off. These channels provide a connection to the bearing surface in a manner such as to cause automatic lubrication. The process is self-starting whenever the oil level is high enough and when the cross channels are tightly sealed.

Another such "sucking" narrow bearing is shown in Fig. 15(B). This is a simpler design and suited only for one direction of rotation. Satisfactory results have been obtained in preliminary tests of narrow bearing with a b/d ratio of 0.8, although the clearance was as great as 0.3 percent.

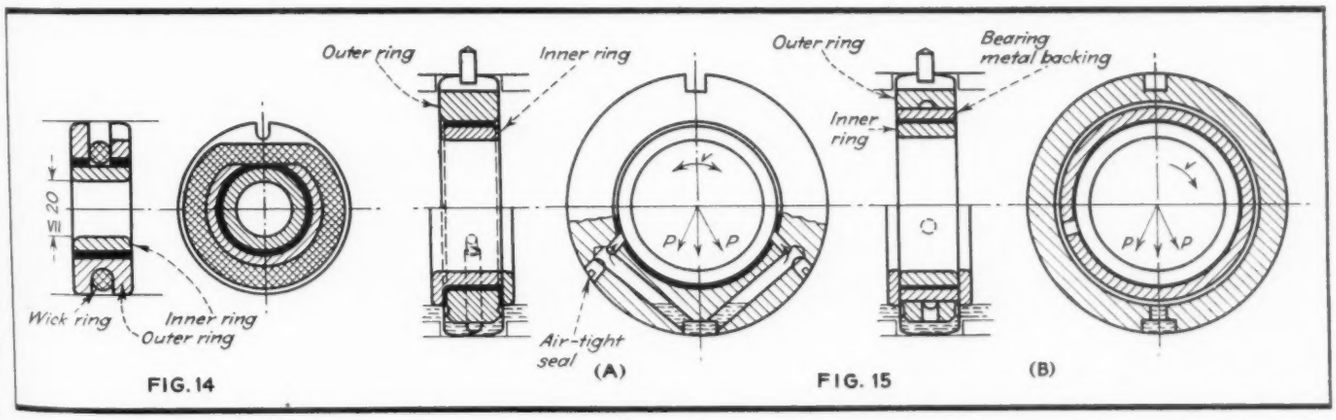


Fig. 14—Small bearing with wick oiling. Fig. 15—Two-ring SMT bearing with vacuum lubrication.

Hard Soldering of Steel

Foreign Abstract condensed from "Hard Soldering of Steel in Protective Gas Atmosphere Inside of Endless Conveyor Furnace" in *VDI Zeitschrift*, October 16, 1943, page 661.

HARD SOLDERING in a protective gas atmosphere is effective in the assembly of small steel parts into steel units. Whether high tensile strength, accuracy in dimensions and a smooth finish are demanded, hard soldering offers the advantage over welding in that the parts are uniformly heated thus avoiding temperature stresses and distortion.

For hard soldering in a protective gas atmosphere, conveyor type furnaces generally are used. The usual conveyor furnace has an electrically heated soldering chamber about 2 meters long. Attached to it is a cooling channel 4 to 6 meters long. A motor driven conveyor carries parts to be soldered into the furnace. Motor speed regulation permits the conveyor travel to be varied between 70 and 350 mm. per minute.

Parts within the soldering chamber must be brought to a temperature above that of

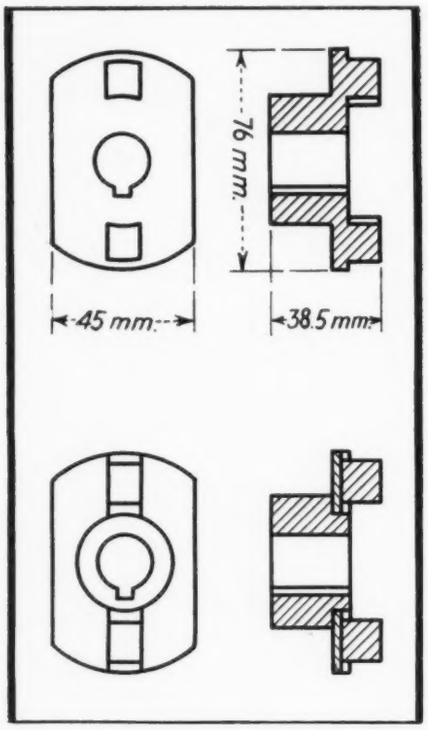


Fig. 1—Fabrication of a coupling by hard soldering instead of cutting it from the solid. Material saving, 45 percent; cost saving, 30 percent.

the solder. Copper having a melting temperature of 1,083 deg. C. is used for hard soldering of steel. Accordingly the electric furnace is adjusted to reach a "glow" temperature of 900 deg. C. in the first third 1,080 deg. C. in the second third; 1,120 deg. C. in the last third. Cooling channel is water-cooled and inside it the parts under a mantle of protective gas cool 100 to 150 deg. C. in order that they will not temper. The furnace, 200 mm. wide, 100 mm. high consumes about 30 kw. Between 6 and 8 cu. meters of illuminating gas and 0.5 to 1 cu. meter of water are consumed each hour.

The protective gas atmosphere is formed by incompletely burning illuminating gas with a fuel-to-air ratio of 1:2, and then cooling, desulphurizing and drying the products of combustion. The gases still contain hydrogen and carbon monoxide. The protective gases stream through the furnace and are drawn off at the exit.

The solder groove must be correctly shaped so that the copper solder is drawn into it by capillary action. This limits the width of the groove to 0.3 mm. wide. Copper has a strong tendency to creep along surface grooves and ridges and this tendency can be utilized to direct the flow of copper only to the desired location. The tearing away strength of flat soldered joints of 0.3 mm. thickness equals tensile strength of copper (18 kg. per sq. mm.) whereas the soldered joint strength of solder thicknesses between 0.2 and 0.3 mm. lies beyond the yield point of the steel. Small parts that are to be soldered together must be held securely in place either by a weight, pressing, grooving, notching or spot welding.

The following metals can be soldered with copper: Carbon steels, low alloy steels, cast steel and malleable iron. Steels with a higher chromium, molybdenum or silica content require protective gases that are free from carbon monoxide or require the use of a flux such as borax. Non-ferrous metals are seldom hard soldered. Cold-rolled steel and other metals lose strength gained through cold rolling when the attempt is made to hard solder them.

Because of the high melting point of copper, hard soldered steel parts can be heat-treated to improve surface hardness.

The principal application of hard soldering is in the fabrication of small complicated parts that were formerly cut out of solid metal with considerable waste of labor and material. These parts are now assembled from smaller individual parts.

Materials for Precision Parts

Foreign Abstract, condensed from "Metallurgical Research Problems in the Precision Parts Industry" by H. Lupfert in *VDI Zeitschrift*, Aug. 7, 1943, page 481.

THE PRECISION PARTS INDUSTRY is concerned with the design of small, accurate parts, such as are built into watches, measuring instruments, office machines, telephone equipment, and sewing machines, and with the manufacture of them in great quantities. The discussion treats the progress made in the use for precision parts of cold drawn steels, thin steel sheet, spring steel, alloy steels, aluminum alloys, zinc alloys and bearing materials.

Sheet steel in the form of strips up to 0.20 in. in thickness and in the form of plates up to 0.15 in. in thickness can be case hardened for wear resistance without materially reducing the impact strength, if the depth of case does not exceed 20 percent of the section thickness. The carbon content of the core is limited to 0.12 percent.

Curve a, Fig. 1, in which the hardness is nearly uniform for about half the depth of penetration and then decreases rapidly toward the core, is the desirable condition. Curve *b* is undesirable, not only because the hardness is to low but also because the hardness decreases at a constant rate.

Spring steels are used in the small parts industry mostly for storing energy, unalloyed

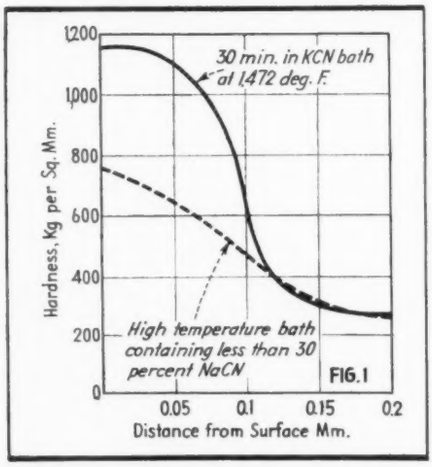


Fig. 1—Sheet steel can be case hardened in baths of either KCN or NaCN and quenched in oil.

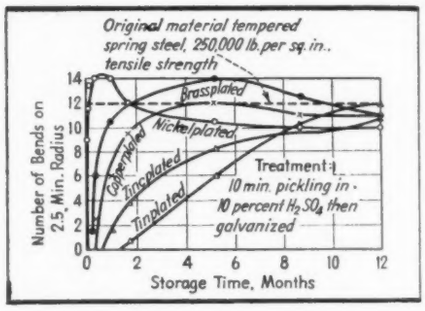


Fig. 2—Variation in flectural number of 0.012 x 0.45 in. spring strip with storage time after pickling and galvanizing.

steels of 140,000 to 280,000 lb. per sq. in. tensile strength being common for flat springs. Corrosion is a problem of the metallurgist. Plating prevents corrosion but has the danger of hydrogen embrittlement. Fig. 2 shows the effect of hydrogen embrittlement resulting from different types of plating and rates of recovery after plating. The nickel, brass and copper plated strips recovered faster than the zinc and tin plated strips. [The article does not attribute the hydrogen embrittlement to the pickling prior to plating.]

Recovery is speeded up by heating. Springs that required a year for recovery at room temperature required only five hours at 200 deg. F., one hour at 300 deg. F, and 20 min. at 420 deg. F. Fully dependable recovery is not achieved.

Since the direction of rolling influences the behavior of alloy steels of high tensile strengths, 213,000 to 240,000 lb. per sq. in., a method was developed for testing small rods taken from the longitudinal and transverse directions to rolling. Quality is judged by the transverse, or weaker specimens. Chromium vanadium steel can be substituted for chromium molybdenum vanadium steel. Shot blasting improves the fatigue life of precision parts, particularly those made from materials of high tensile strength.

The dimensions of zinc alloy parts change even at moderate loads. Zinc alloys also have low impact strength, especially at temperatures of 100 deg. F. below zero. Low friction bearings are made of plastics, hardened steel and aluminum alloys. Thirty-nine references are listed in an appendix.

Measuring Surface Roughness

Foreign Abstract condensed from "New Methods for Measuring Surface Roughness", in *VDI Zeitschrift*, February 20, 1945, page 114

MASS PRODUCTION of high quality interchangeable parts has increased the demand for a dependable method of detecting and measuring surface imperfections. Most recent method for surface measurements bear a strong resemblance to the known feeler gage or prismatic light gage devices. These gages are too sensitive for many shop applications for they fail to come up to the requirements of high accuracy. The problem has now been tackled photometrically. A photometer is used, first to measure the light reflected from the investigated surface when illuminated as a bright field, and then to remeasure the light reflected when the surface is illuminated from all sides as a dark field. The ratio obtained from these two readings may be regarded as an index of the relative quality of the surface. By adopting dark field illumination from all sides, the process is independent from the directional position of the surface irregularities as long as the meter points vertically to the surface being inspected.

The equipment used is shown in Fig. 1. It consists basically of a microscope with an adjustable lighting shutter, a translucent intermediate blender, and a light concentrating lamp. Part of the light reflected by the roughness of the surface enters the binocular assembly and reaches the eye piece, which is used to properly focus the instrument to view the given surface area. The remaining part of the reflected light is picked by photocell from which the resulting current output is measured with a milliameter and a stop watch. Repeated measurements with both bright and dark field illumination average the fluctuation in the intenslity of the light source.

This technique is useful on cylindrical and flat surfaces that show a surface variation ranging between 20μ to under 0.5μ. It has been found that repeated measurements taken on one given test spot differ very little, and also that a comparison between different test areas on the same surface of the part show little variation. All observations seem to indicate that the values obtained by optical methods bear a direct and close relationship to the absolute roughness, which proves the true worth of the method.

The method has been used in the inspection of high quality precision tools such as test gages and plates, and it is also useful in inspecting lapped and polished flat surfaces. It has been noted that frequently the same test points repeatedly gave the same numerical results. Accuracy of this order could not always be obtained either by the prismatic light method, or by feeler test equipment having an enlarging ratio of 1:2,000.

The method is distinctly superior when applied to surface roughnesses under 1μ, but it has other advantages, such as immunity to vibration because there is no material contact of parts during the relatively short measuring time. Also it is feasible to enlarge the viewed surface to the desired degree. Recently it has been discovered that the method is applicable to arched surfaces providing that the arch curvature does not exceed 2 mm. in height.

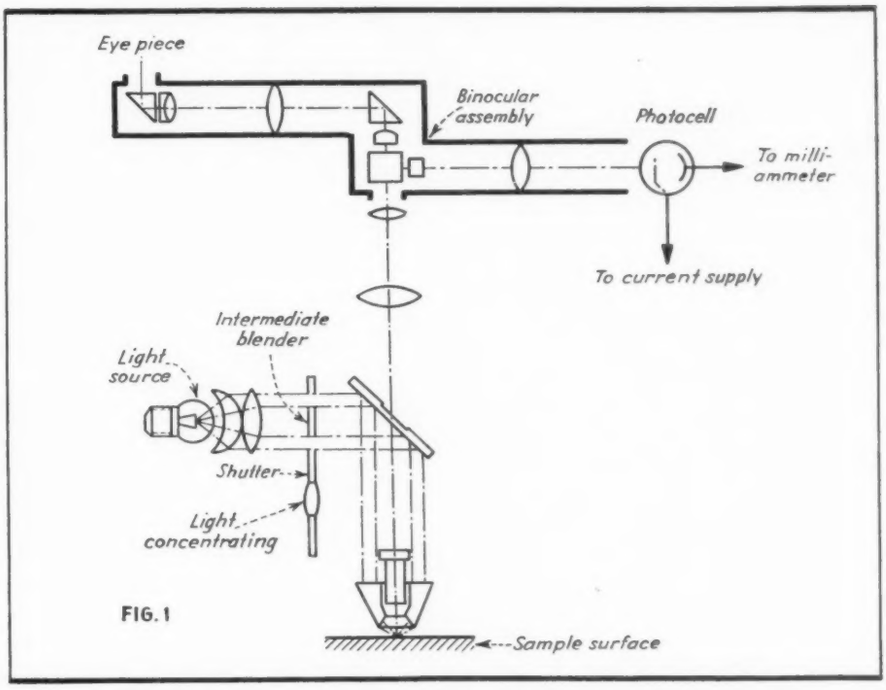


FIG. 1

Fig. 1—Schematic arrangement of surface inspection apparatus.

Bolt Bearing Strength

From "Bolt Bearing Strength of Wood and Modified Wood" in Fasteners, Vol. 2, No. 4, published by American Institute of Bolt, Nut and Rivet Manufacturers.

STUDIES at the Forests Products Laboratory of the United States Department of Agriculture, in cooperation with the University of Wisconsin, revealed that the method of drilling bolt holes in wood greatly influences its bolt bearing strength. The type of drill used and the rate of feed when drilling are two of the more important factors that determine bolt bearing strength.

Analysis of 259 bolt holes covered by Report Number 1523 prepared by H. R. Goodell and R. S. Phillips indicate that (1) bolt holes with visibly smooth side walls have bolt bearing properties far superior to those with visibly rough side walls; (2) to produce a smooth hole, the drill must be well sharpened and the rate of feed in drilling must be slow enough to allow the drill to cut rather than tear its way through the piece; and (3) in Douglasfir plywood and Sitka spruce, a twist drill produces a smoother hole than does a machine bit.

Unless holes are properly drilled, the bolt bearing loads allowed in the ANC Design Handbook cannot be sustained. For poorly drilled holes, the stress at the proportional limit may be as low as 1/3 the allowable limit stress thus computed. The variables introduced by poorly drilled holes hardly permits the use of the common design assumption that the load on multiple bolted connection is equally divided among the bolts.

Silver Alloy Brazing

From "Silver Alloy Brazing in the Manufacture of Instruments" by Adolph Bregman, in *The Instrument Maker*, Nov.-Dec. 1945, published by The Instruments Publishing Company, 1117 Wolfendale St., Pittsburgh.

REASONS for the growth of silver alloy brazing are (1) the development of new brazing alloys that have low melting temperatures, high ductility, good resistance to corrosion, high strength, high electrical and thermal conductivity and that are free flowing: (2) the use of these alloys as inserts, pre-placed in the joint before heating; (3) suitability to many kinds of heating; (4) high speed of operation and low cost of production, resulting mainly from low flow temperatures and fast brazing action.

Heating methods in general use are: (1) The oxy-acetylene hand-operated torch; (2) the gas or electric furnace, either of the batch or conveyor type; (3) high frequency induction heating; (4) gas-air burners.

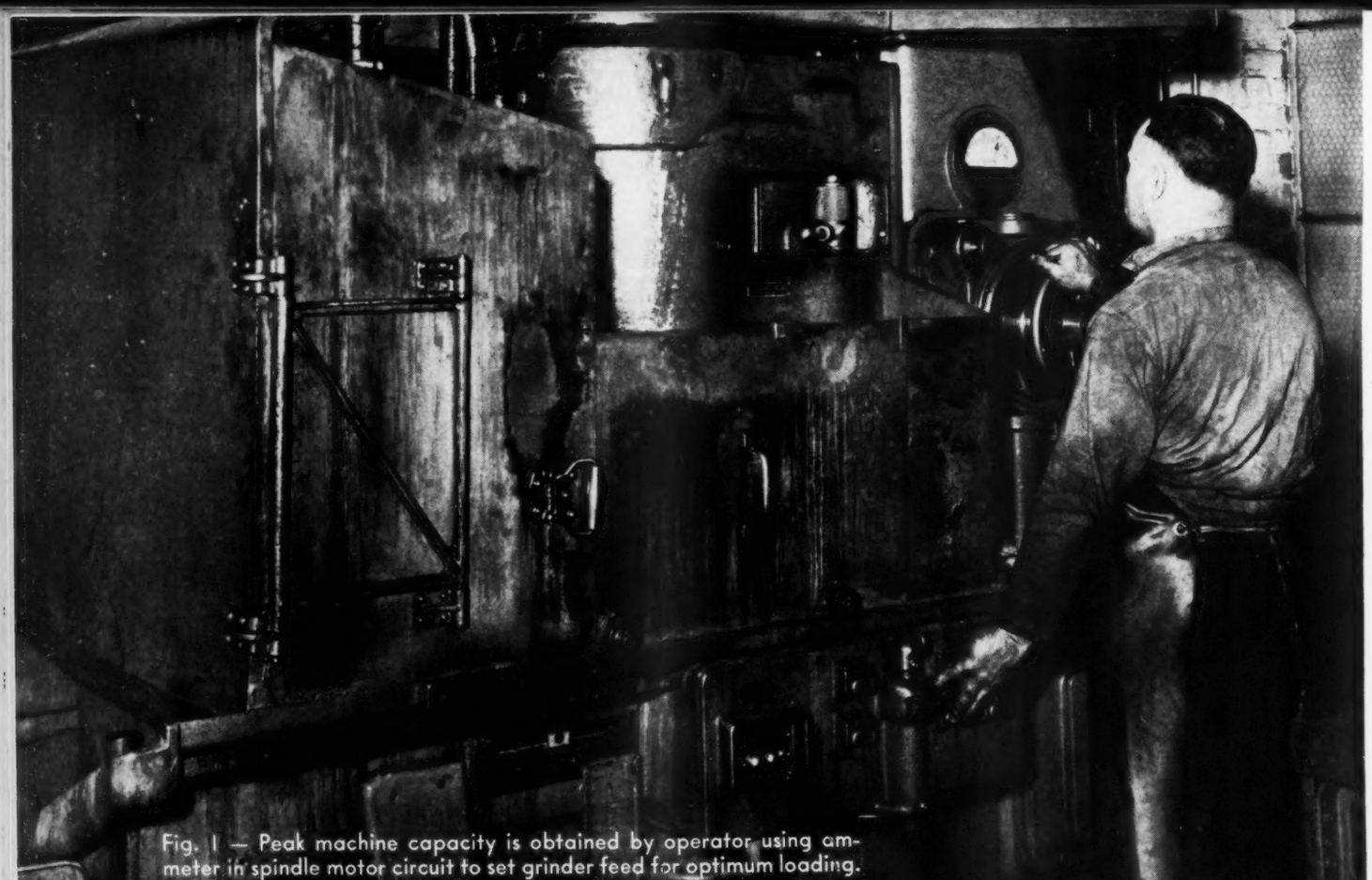


Fig. 1 — Peak machine capacity is obtained by operator using ammeter in spindle motor circuit to set grinder feed for optimum loading.

Blanchard Machine Company

Instruments Often Help Machines Work at Peak Efficiency

JOHN H. MILLER
Chief Engineer, Weston Electrical Instrument Corporation

Typical functional uses of instruments on grinders, die sinkers, and automatic screw machines for maximum loading or optimum operation. Simplified method of metering a variety of motors that differ in horsepower, voltage, or phase by using a single standard meter matched to the load by an adjustable doughnut transformer is discussed.

ELECTRICALLY driven machines are increasingly using built-in ammeters and wattmeters to help the operator maintain rated full capacity output. When several motors are integrated into a single machine, such instruments are especially valuable in helping the operator coordinate the best individual loads for overall optimum operation.

Because of their diversified problems, machine tools often find electrical instruments particularly useful as an operating tool. For example, heavy duty surface grinders commonly use a separate motor directly driving the grinding wheel as shown in Fig. 1. There are adjustments for various depths of cut but the only criterion for determining the maximum cut is the power taken to drive the wheel. Presumably this should equal the motor rating; or, if the wheel itself is the limitation, some lesser value. An ammeter or a wattmeter, scaled in amperes, watts, or horsepower, and indicating the power taken, will show when depth of cut is correct for maximum removal of metal under limitation of motor or wheel maximum values.

For example, for the same maximum power consumption a deeper depth of cut is required for grinding cast iron flat-iron bases than when grinding planer knives. Although a mechanical power or torque meter could be used to indicate maximum power, the ammeter or wattmeter method is so simple that it is usually chosen for electric drives.

Instrumentation is not only useful for maximum loading but also for optimum loading. For example, a light cut sometimes not only reduces production but also causes excessive and needless maintenance. To avoid both these hazards the surface grinder in Fig. 1 uses an ammeter in the spindle motor circuit to indicate the right grinding feed and to show if the wheel requires dressing.

Optimum Operation

When grinding on the full surface of the work the load on the spindle motor is ordinarily full load or more. Under the usual condition of using a wheel suitable for the full power of the machine, if too little power is used because of a shallow cut the wheel may glaze and refuse to cut. The ammeter guides the operator in adjusting rate of grinding to the correct feed and if his setting is incorrect he is warned by the increasing power that is consumed as the wheel begins to quickly glaze or dull. A wheel that is correctly matched to the work wears away fast enough to keep itself sharp and never requires dressing. However, even with the machine adjusted to the proper power consumption, if the wheel used is a little too hard it will gradually dull and require dressing. As the wheel slowly glazes, the corresponding increase in the ammeter reading lets an operator determine the most economical point in the operating cycle to redress the wheel for best overall production.

Instruments are also often used for optimum loading in mass production automatic machinery. For example, in operating heavy automatic screw machines there is always the problem of deciding at what point the tools should be resharpened or replaced. With a fixed cycle of tool feed for a given cut, the power and current increase as the tools become dull. On a given cut, experience dictates the optimum point for tool replacement. Once determined, the condition can be read on the meter, the machine shut down, and tools resharpened or replaced. Maximum production of uniform parts is secured with minimum rejections because of dull tools.

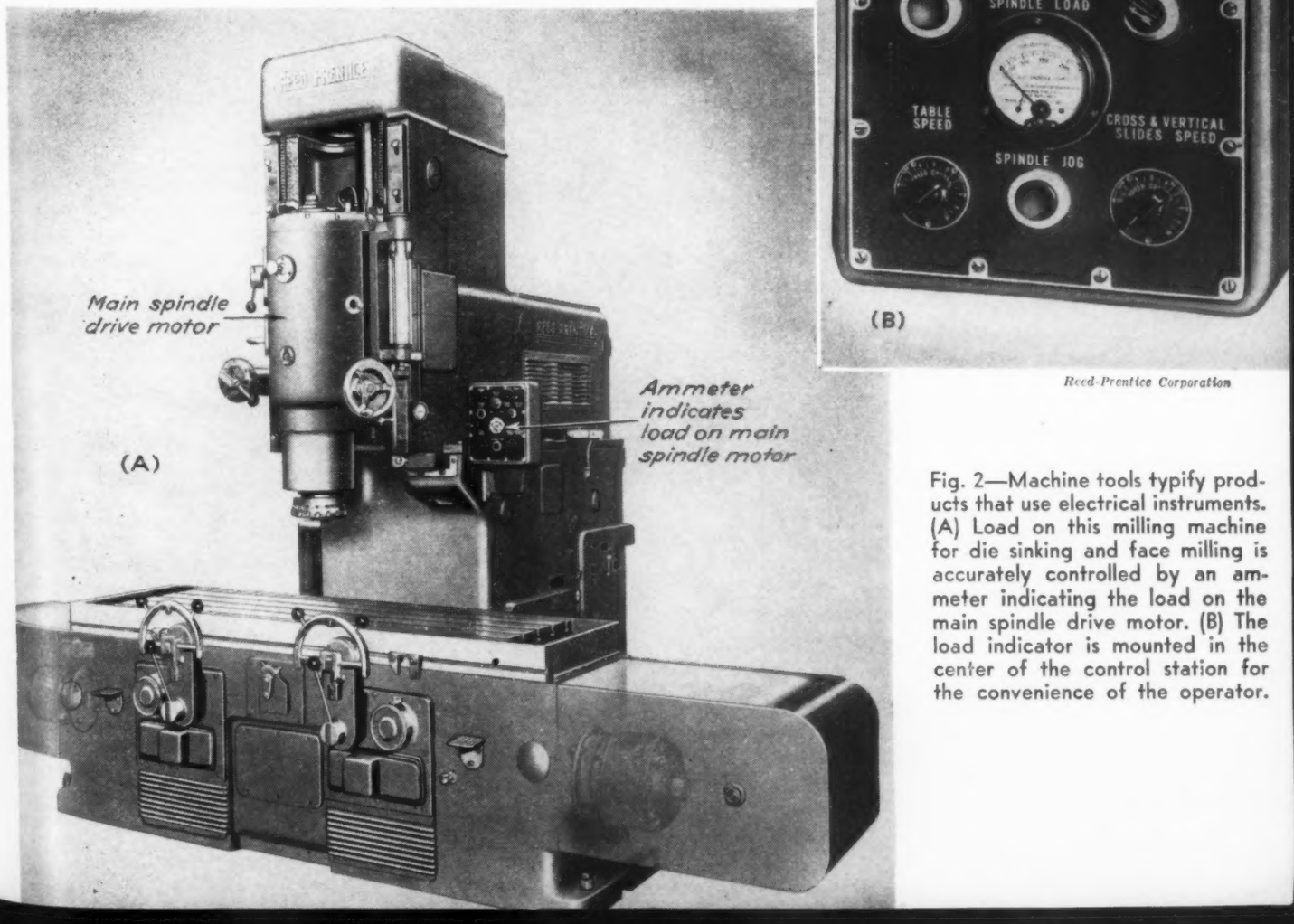
Another example of intelligent instrumentation, is the vertical milling machine shown in Fig. 2, that was designed especially for die sinking and face milling. An ammeter tells the operator at all times the load on the main spindle motor.

Many operators have little idea of the load they can place on a machine and so they rarely use it to capacity in order to be on the safe side. Also the instrument has proved to be good protection against overloading the machine. Although the machine has standard motor overload protection, when the overload relay stops the motor it usually breaks the cutter. This is particularly true when tungsten carbide cutters are used, therefore is is important that the motor is not stalled.

Although either a wattmeter or an ammeter can be used on machines they are quite different instruments. The wattmeter costs about twice as much as an ammeter, and is a more complex mechanism though not in any sense fragile. Its readings represent true power to the motor, and, if the no-load motor watts are subtracted, the remainder is almost exactly proportional to power absorbed by the machine. In contrast, the ammeter measures only current, and is not an accurate measure of power at low loads because of the influence of wattless current.

Power Factor

On certain types of a.c. induction motors the current reading increases only moderately from no load to full load; it is actually the power factor that changes from as low as 20 percent at no load to approaching unity power factor at full load. Because of power factor, the use of an ammeter to measure power must be considered in terms of the characteristics of the motor used. However, an ammeter is an extremely satisfactory instrument to prevent motor overloads. Beyond the value of representing full



Reed-Prentice Corporation

Fig. 2—Machine tools typify products that use electrical instruments. (A) Load on this milling machine for die sinking and face milling is accurately controlled by an ammeter indicating the load on the main spindle drive motor. (B) The load indicator is mounted in the center of the control station for the convenience of the operator.

load, the upper section of many ammeter scales are printed in red or otherwise marked to indicate an overload condition.

Meter Visible

To be useful, the meter must be visible to the operator and is usually mounted on the machine at some distance from the motor. This means that heavy cables must carry full motor current to the instrument, unless current transformers are used on alternating current or shunts on direct current. For a.c. motor drives, the transformer can be placed in the line at the motor or between it and the contactor, and relatively light No. 14 wires run to the meter, as shown in Fig. 3. Often the current transformer is mounted inside the control cabinet. See Fig. 4.

Also, one of the transformer secondary leads can be grounded so that high voltage insulation is not a problem in the meter wiring. For these reasons, transformers are being used for moderate powers of 5 hp. and greater.

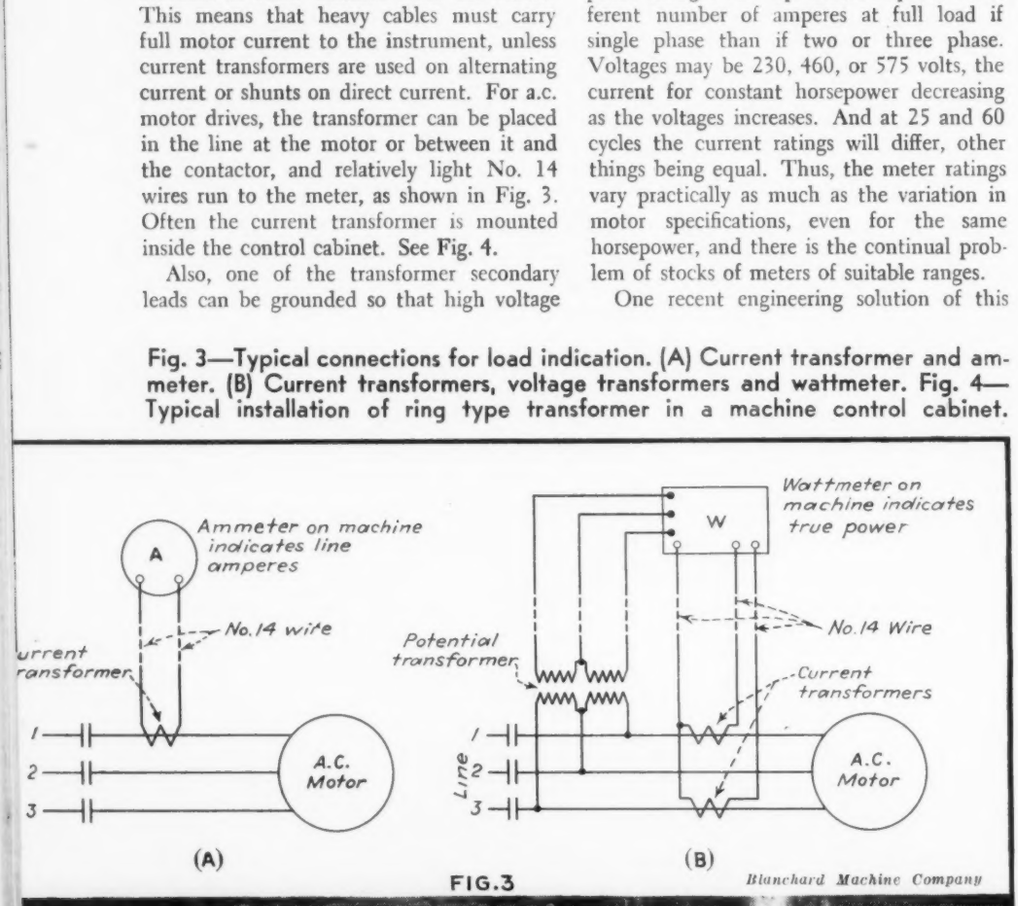
In the application of an ammeter or a wattmeter to a machine tool having a given size motor, there is the problem of the meter rating as the power line voltage and phase change. A 5 hp. motor requires a different number of amperes at full load if single phase than if two or three phase. Voltages may be 230, 460, or 575 volts, the current for constant horsepower decreasing as the voltages increases. And at 25 and 60 cycles the current ratings will differ, other things being equal. Thus, the meter ratings vary practically as much as the variation in motor specifications, even for the same horsepower, and there is the continual problem of stocks of meters of suitable ranges.

One recent engineering solution of this motor problem is to apply identical meters having the same current coil rating and then match the meter to the load through a suitable current transformer.

For these meters the scale is usually in percent, such as "Motor Load-Percent," "Percent Full Load," "Horsepower, %," such as shown in Fig. 5. The scale is usually marked for 125 or 150 percent load depending on the overload that the machine designer wishes. Even though the scale is red above 100 percent, or marked "Danger," experience indicates that as long as the meter pointer does not pass full scale, some operators will assume no harm can be done. But when the pointer is off scale, there is no question in the operator's mind but that the system is overloaded.

The safe allowable overload shown on the scale and which the motor will carry for a short period without damage, depends on the electrical and thermal overload capacity of the motor and the operating cycle of the machine.

Fig. 3—Typical connections for load indication. (A) Current transformer and ammeter. (B) Current transformers, voltage transformers and wattmeter. Fig. 4—Typical installation of ring type transformer in a machine control cabinet.



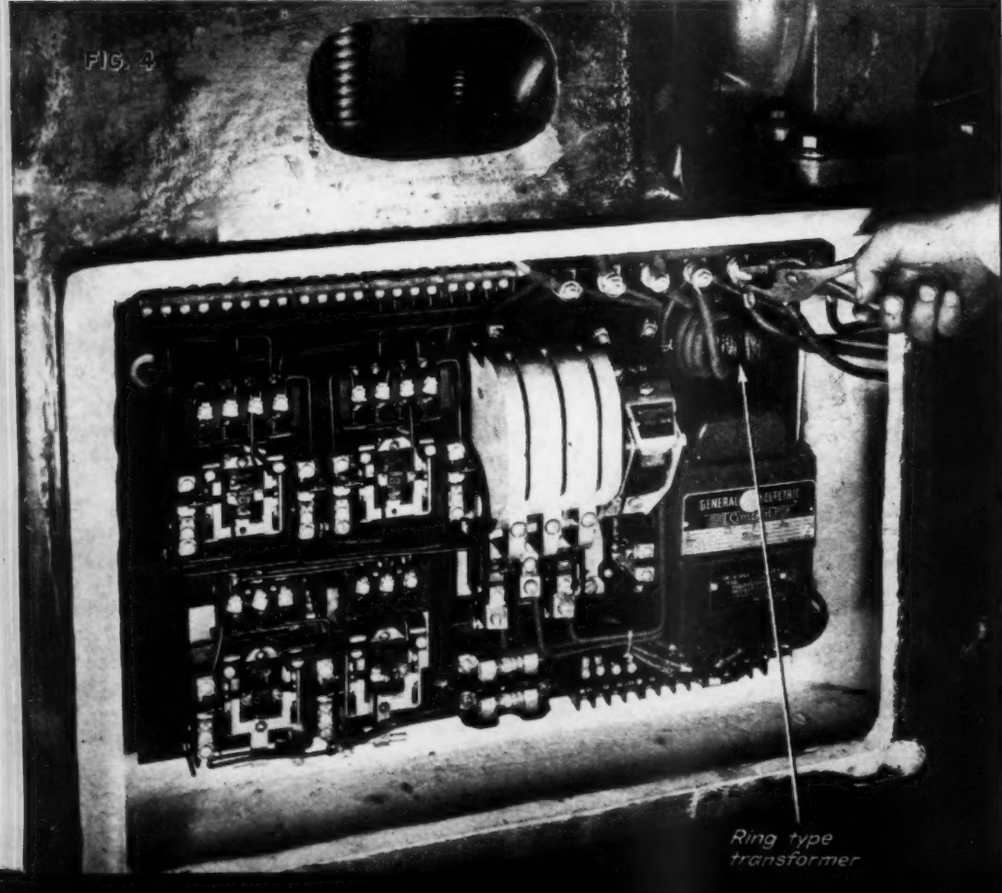


Fig. 5—For simplicity in using a single instrument as a load indicator, scales are usually marked in percentages. Danger zone is shown by colored section for guidance of operator.

Adjustable Transformer Ratio

The single instrument for all motor ratings and sizes is made practicable by a single transformer that can be readily matched to the particular range of requirements for various motors. The transformer is a doughnut or ring type transformer with a range near the highest current to be measured. For lower currents 2 or 3 turns of one of the motor leads are placed through the ring to give one-half or one-third of the original transformer primary rating. For additional flexibility and to provide the intermediate values the secondary winding terminates in long flexible leads so that additional turns can be added. If fewer secondary turns are required, then turns can be wound in re-

verse, which effectively subtract from the main winding, as shown in Fig. 6. In other words, the long leads provide a vernier adjustment for matching the transformer ratio to a range of currents. Ampere turns in the primary must equal those in the secondary.

For example, assume a 200 to 5 transformer, supplying a 5 amp. meter. To obtain full scale on the ammeter with 200 amp. flowing in the single turn primary, the secondary will have 40 turns, disregarding a turn or a fractional turn for compensation. For 100 amp. or half the previous line current, the primary requires 2 primary turns to produce the necessary 5 amp. to give full scale reading on the ammeter.

If the meter is to read full scale when 125 amp. flow in motor circuit two turns are wound on the primary and the necessary 50 secondary turns are obtained by adding 10 turns with the flexible-secondary leads. Actually, five turns are wound with each lead in the direction of the original winding. For full scale with 120 amp., two primary turns are used and eight turns are added; for 122.5 amp., nine turns are added. By this method of threading the right number of turns through a ring transformer, a single instrument can be matched to a current range up to 200 or 250 amp. with less than two percent error. Stocking a single instrument and a single transformer purchased in quantity, provides a substantial saving.

Simple Table

For simplicity and accuracy, a tabulation such as Table I is the usual way to show just how many turns to add or subtract. Similarly, tables for other horsepowers can be computed up to the current limit of the transformer. For example, a complete line of oil pressing machines requiring from 5 to 40 hp. motors and using a single size of meter and transformer, was tabulated for the various motors at all voltages to 550 volts, two and three phase.

Since many of the applications are for overload indication, allowed error of two percent does not require precise transformer compensation. Also, normal voltage variations will make the translation from power to current somewhat approximate with ammeters, but the simplicity of the adjusted transformer method and its economic advantages seems to justify its use. Overloads of damaging degree are definitely indicated, and the accuracy is ample for many simple loads.

Also, the method can be used with a wattmeter for more exact power readings and precision can be secured at extra expense.

Although most industrial machines are a.c. a similar method can be used for d.c. motors: For example, instrumentation may be especially applicable for machines using d.c. adjustable-voltage control for obtaining maximum production throughout a wide range of speed. On such d.c. applications, a single 50 millivolt d.c. meter with a percentage scale is used with specially adjusted shunts.

Instrumentation of machines will probably increase in the year or two ahead as industry swings into full scale peacetime production and low unit costs become increasingly significant. Also, by the proper integration of instruments into industrial machinery, designers can often substantially help the operator consistently utilize the optimum capacity that was so carefully designed into the machine. This produces maximum output with safety. The method of using a standard five ampere a.c. ammeter with a percentage scale and an adjustable transformer will definitely help solve one of the industry's oldest problems, a simple and inexpensive way to control a.c. motorized machines.

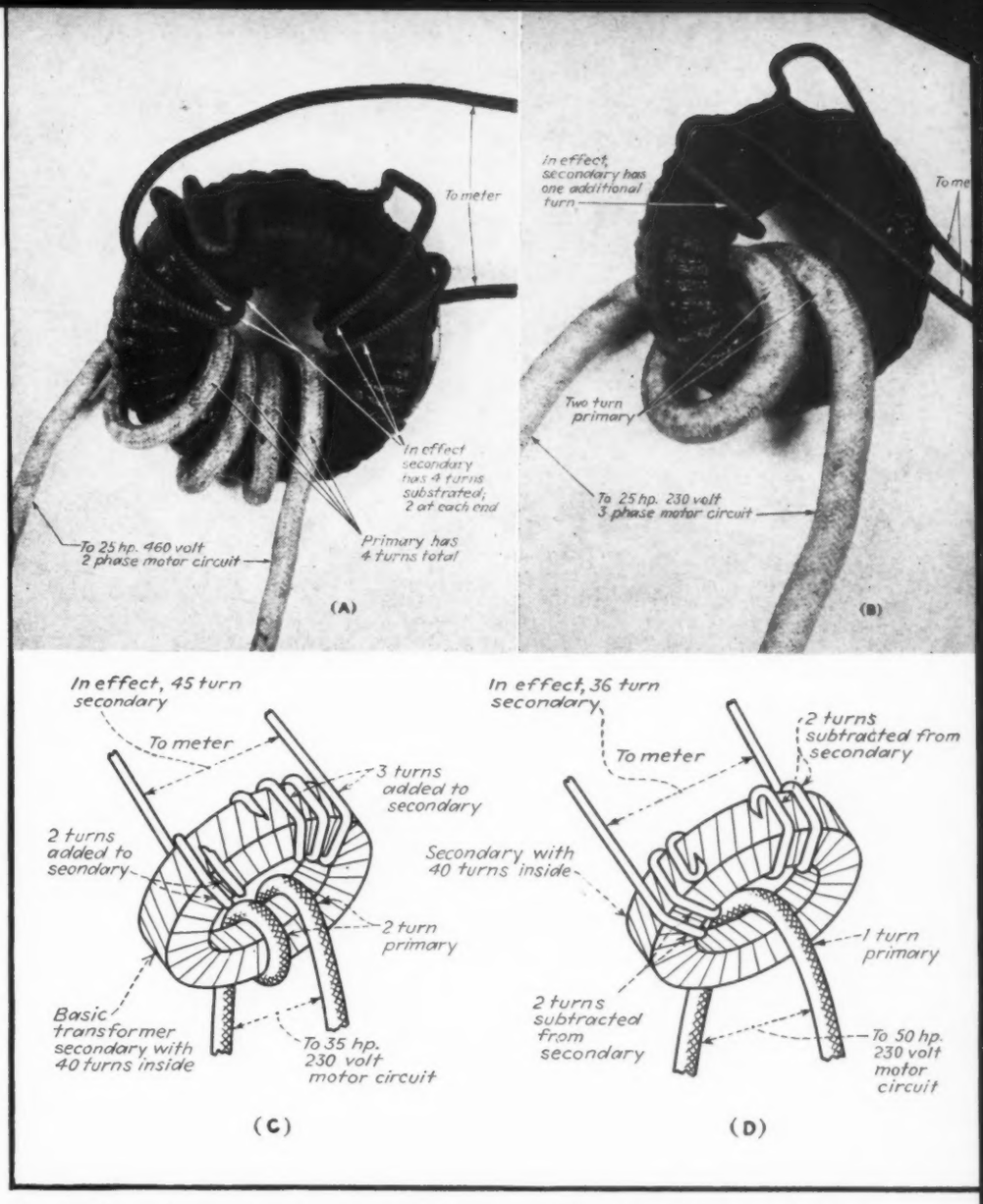

To meter
In effect secondary has 4 turns substrated; 2 at each end
To 25 hp. 460 volt 2 phase motor circuit
Primary has 4 turns total
(A)
In effect, secondary has one additional turn
To me
Two turn primary
To 25 hp. 230 volt 3 phase motor circuit
(B)
In effect, 45 turn secondary
To meter
3 turns added to secondary
2 turns added to seondary
2 turn primary
Basic transformer secondary with 40 turns inside
To 35 hp. 230 volt motor circuit
(C)
In effect, 36 turn secondary
To meter
2 turns subtracted from secondary
Secondary with 40 turns inside
1 turn primary
2 turns subtracted from secondary
To 50 hp. 230 volt motor circuit
(D)

Fig. 6—A special ring type current transformer provides adjustable ratios for the single rated instrument. (A) Doughnut transformer for 25 hp. 460 volt, two-phase motor of Table I; only turns that pass through hole are effective. (B) Similar ring transformer for 25 hp., 230 volt, three-phase motor. (C) Five turns added for metering 35 hp. 230 volt two-phase motor. (D) Four turns subtracted for metering 50 hp. 230 volt three-phase motor.

Table I—Variations of 200/5 Normal Transformer Ratio for 25 Hp.

Voltage	Phase	100% Load, Amp.	150% Load, Amp.	Pri. Turns	Sec. Turns
230	3	68	102	2	+1
230	2	60	90	2	−4
460	3	34	51	4	+1
460	2	30	45	4	−4
575	3	27.2	40.8	5	+1
575	2	24	36	5	−4

Lightweight Electrical Systems Design Problems and Solutions

T. B. HOLLIDAY*
President, Mueller Accessories, Inc.

Aircraft electrical systems are described and design practices compared with those used in the development of industrial electrical equipment. Effect of voltage on the weight of wiring used in aircraft is discussed.

THIS ARTICLE is devoted to reduction of weight in the aircraft electrical system. This system is described as a 24 to 30 volt d.c. system and consists of variable speed generators mounted on the main engines with voltage regulators set at 28 volts. These are operated in parallel with 24 volt batteries. The 30 volt rating of the generator gives some margin of performance over the 28 volt setting of the voltage regulator. The battery is a 12 cell lead acid type quite similar to automotive and truck batteries in construction.

The power supply bus is distributed throughout the airplane. In multi-engine aircraft a transverse bus connects the outboard generators to a longitudinal bus between the nose and tail. Circuits to individual equipment from this power bus are protected by circuit breakers.

Weight Distribution

The distribution of weight in three sizes of military aircraft is shown by Fig. 1. As the size of the airplane increases there is a definite increase in the portion devoted to structure, but at the same time equipment weight drops off. The same type of equipment will often suffice regardless of airplane size and, therefore its penalty is less in larger aircraft. The increase in structural weight permits a greater fuel and pay load, thereby giving greater range. Armament would occupy a large share of this weight in military aircraft. Electrical equipment and furnishings show a tendency to rise with aircraft size.

The proportionate weight of electrical power equipment goes down with size. Yet the generating capacity of the small airplane is 6 kw. and in the largest, it is 54 kw. The proportion devoted to wiring increases with size. This can be partially explained by the low system voltage. The 24 to 30 volt system is not an optimum value for aircraft being built today. The portion devoted to motors is difficult to evaluate since they are usually integral parts of other equipment. The estimates as given in Table I include gear reductions, but not driven accessories.

Reduction in weight is a problem in design. The combined problem of obtaining peak efficiency, long life, reliability and minimum weight is the greatest challenge of design in engineering. The primary problem is to obtain more output per cu. in., and, therefore, per lb. of material. It is imperative that the best possible materials be used. While the cost per lb. of material will rise, the amount of material used will decrease. Further, the lowered losses in the better materials will pay dividends in decreasing the amount of cooling which is needed.

The next attack to lower the weight of electrical machinery is to use higher working densities. By using the best grade of iron it is possible to crowd more lines of flux into each sq. in. Each conductor must carry the maximum amount of current. Carbon brushes must carry unprecedented currents. All of these will increase the power loss per lb. of materials but the number of pounds decrease almost as rapidly. The excess must be cared for by improved cooling or operation at higher temperatures.

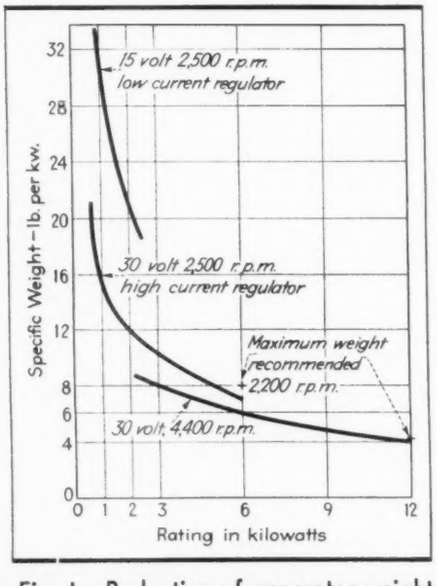


Fig. 1—Reduction of generator weight obtained by changing the type of voltage regulator and by doubling generator speed.

A third attack to reduce weight of rotating machines is to use higher speeds. This will present other and serious mechanical and structural problems such as bearings, rigidity of commutators and construction of armature windings. But speeds up to perhaps 12,000 r.p.m. pay dividends in reducing weight of rotating machinery.

The output of a generator armature can be expressed in terms of its dimensions. Where:

W = output, watts.
Z = number of conductors.
E_c = volts per conductor.
I_c = current per conductor, amp.
N = speed, r.p.m.
φ = flux per pole.
P = number of poles.
B = flux density in the air gap.
L = armature length, in.
D = armature diameter, in.

$$W = ZE_cI_c$$

$$E_c = \frac{\varphi P N}{60} \times 10^{-8}$$

$$\varphi P = BL\pi D$$

$$W = \frac{Z\,BL\pi D\,NI_c}{60}$$

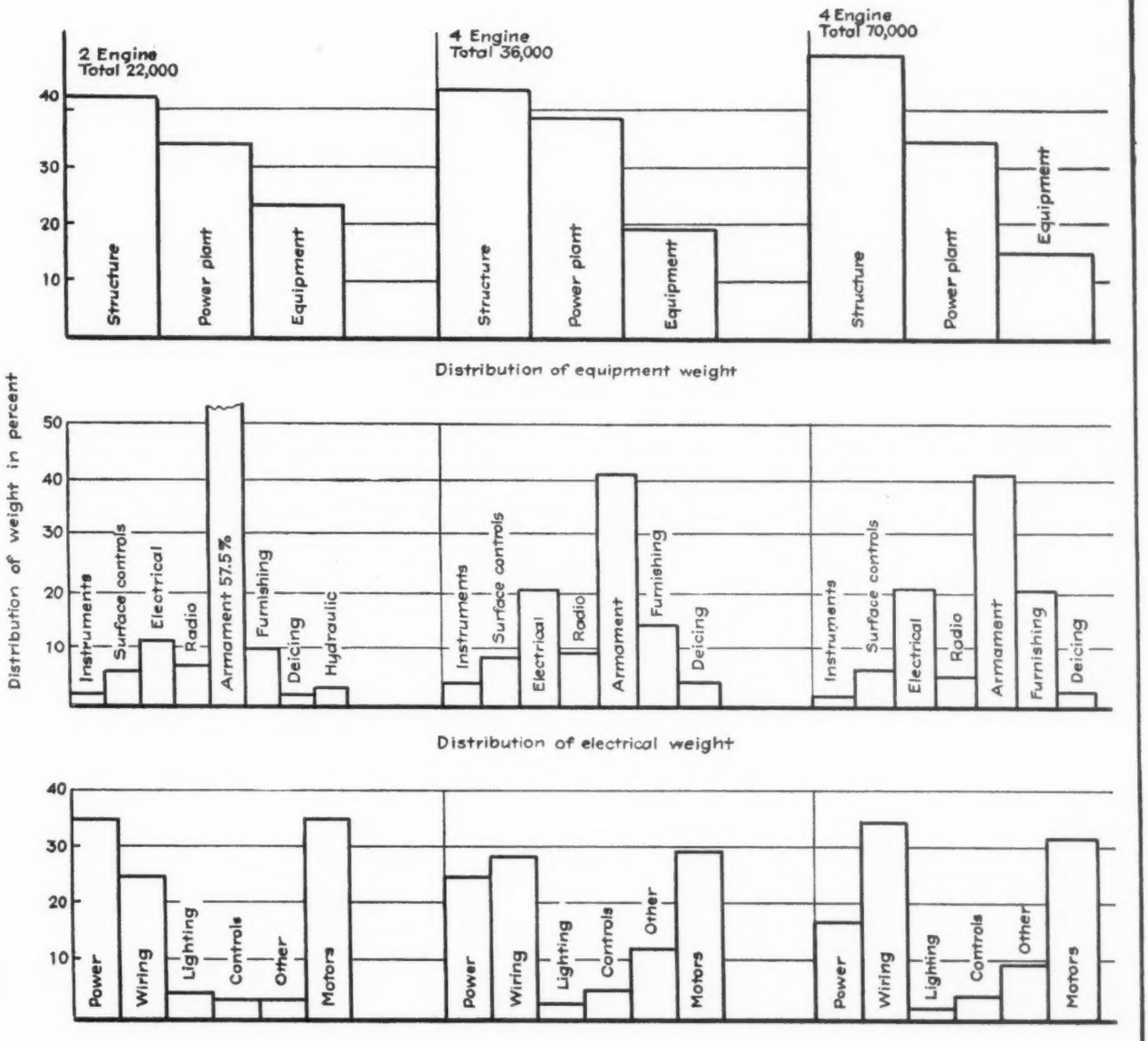
When: $\frac{ZI_c}{D} = q$ and $\frac{60}{\pi^2} = k$

$$LD^2 = \frac{W}{N} \times \frac{k}{Bq}$$

The last equation shows that the volume, and therefore, the weight of the armature is proportional to the output. This volume changes inversely with speed, air gap density, and surface loading measured in terms of amp. conductors per in. of armature periphery. The machine must deliver the required output. To hold weight within minimum values the designer must, therefore, increase speed, increase air gap density or increase the surface loading either by arrange-

* *Formerly Ass't. Chief, Equipment Laboratory, Engineering Div., Air Technical Service Command.*

Table I—Study of Weight Distribution of Three Types of Military Aircraft



ment of conductors or by increasing the current density within the conductors.

Speed produces immediate problems. The improvement, Fig. 1, in specific weight as measured in lb. per kw. was achieved by doubling the rated speed of engine-driven generators. Doubling speed again would not produce comparable results because the mechanical problems of holding an armature together at a maximum speed of 20,000 r.p.m. would tend to offset other gains in weight. There is an optimum speed for any generator or motor rating.

External factors can also affect weight of a given machine. The Tirrell, automotive, type of voltage regulator is limited in both current and voltage. There are critical values of both for even the best contact material. The design problems in 30 volt generators exceeded both limits. Therefore, it was necessary to seek another solution. Improvements in regulators included the multi-contact and the carbon-pile types that gave astonishing improvements in generator performance. The Tirrell regulator is not satisfactory beyond 1.5 amp. and 20 volts. The carbon pile type controls 9 amp. at 30 volts. When it was possible to use this amount of current in the generator field coils it was possible to make the iron portion of the generator do its share of the work. When less iron was needed, the length of conductors in the field coils decreased.

The flux density across the air gap is dependent on the density in iron portions but it is also a second factor that can be used to decrease weight. Comparison of aviation practices with industrial practices is shown by Fig. 2(A). References to industrial prac-

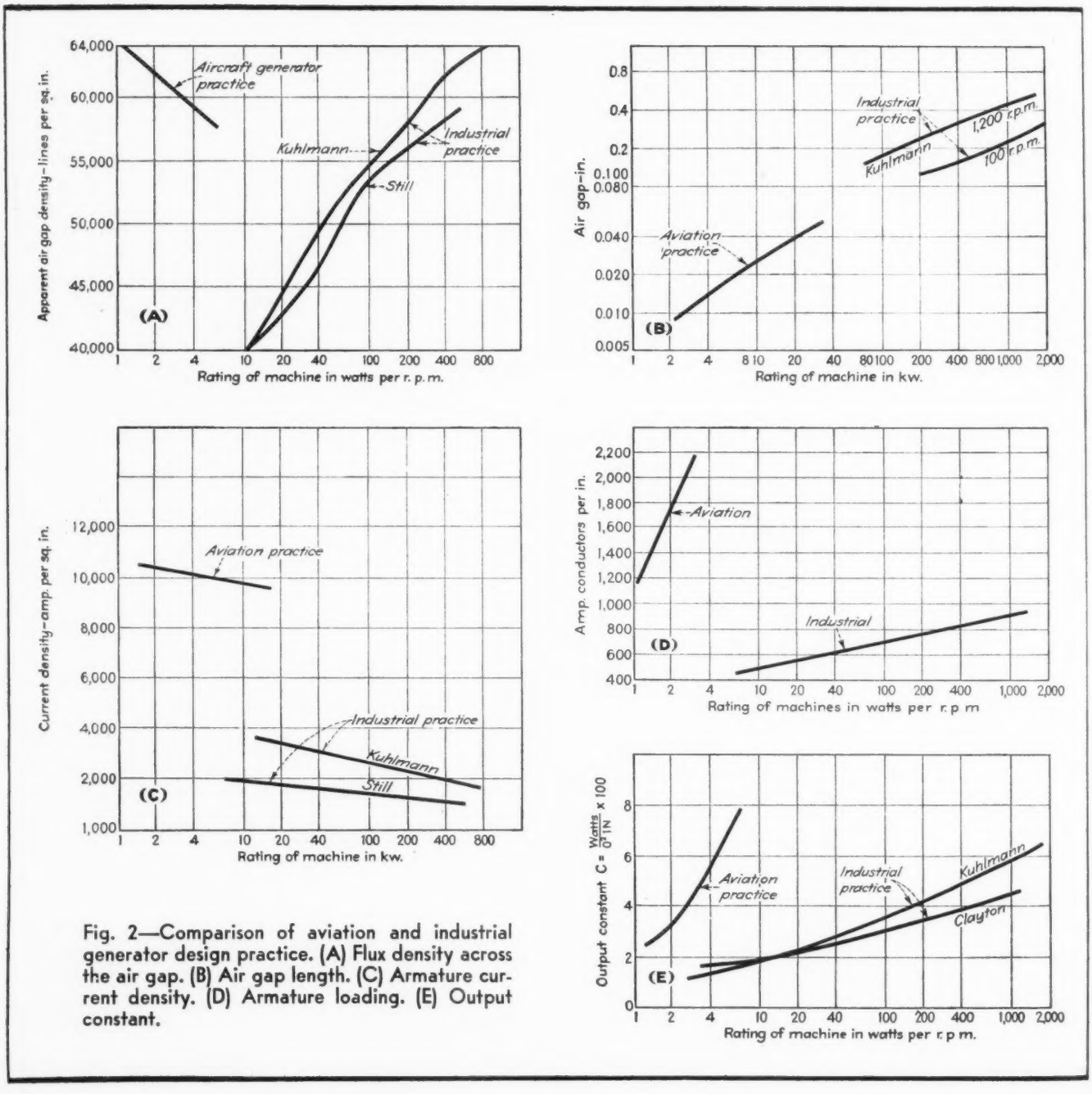


Fig. 2—Comparison of aviation and industrial generator design practice. (A) Flux density across the air gap. (B) Air gap length. (C) Armature current density. (D) Armature loading. (E) Output constant.

tice have been plotted from data given in design texts by Still, Kuhlmann and Clayton.

A comparison of air gaps used in aviation practices and industrially is shown by Fig. 2(*B*). It is obvious that the use of a short air gap is advantageous, since

Where:

φ = flux across the air gap
H = magnetomotive force
g = length of air gap, in.
T = number of turns in magnetic field
I_f = field current, amp.
$\varphi = Hg = TI_fg$

The armature diameter in most rotating machines used in aircraft is less than 4 in: This is because generators have been confined to an external diameter of 6 to 6 3/8 in. In this diameter it has been possible to obtain 12 kw. at 5,000 r.p.m. These small diameters make it possible to use comparatively small air gaps. Since the length of the air gap is a mechanical problem, it is only logical that armatures having a small diameter would require a short air gap.

Minimum Magnetic Reluctance

Use of the best magnetic materials in the armature, poles and yoke, serves to reduce magnetic reluctances, and this in turn will reduce the amp. turns in the field coils. The designer of electrical equipment for aircraft will use materials with little regard to expense. In d.c. machines the use of low loss steels is advantageous only in the armature. In a.c. and d.c. generators the number of poles range from four to eight. Operating speed ranges are 2,500 to 5,000 and 4,400 to 10,000 r.p.m. In each of these the normal operating speed is approximately 3,000 and 6,000 r.p.m. At these speeds frequencies inside the armature will be as follows:

	3,000 r.p.m.	6,000 r.p.m.
4 pole	100	200
6 pole	150	300
8 pole	200	...

The maximum range is 80 to 500 cycles per sec. when including minimum and maximum speeds.

Motors usually have two, four or six poles and operate at higher speeds. Fre-

quencies used in current practices include:

Poles	Speed	Frequency
2	24,000	400
2	15,000	250
4	12,000	400
4	8,000	267

Motors, like generators, show a definite trend toward the 200 and 400 cycles per sec. frequency.

The alternating current system being developed by the Equipment Laboratory, Air Technical Service Command is based on 400 cycles per sec. as being the best overall choice. All of this forces the designer to consider frequencies of three to seven times the commercial 60 cycle per sec. frequency.

Magnetic steels are judged by several characteristics, but they are rated in terms of core loss in watts per lb. with a specified flux density and frequency. They range from "armature" to "special transformer" grades. The armature grade will have three times the core loss of the best transformer grades yet the cost of the best transformer steels do not rise in the same proportion. Therefore, the designer can seriously consider the best steels since the overall cost may be reduced.

The designer must depart from commercial standards. It is advantageous to use thinner sheets of magnetic steels. Where the minimum thickness in industrial practice is usually 0.014 in., sheets of half that thickness have been used in aircraft devices.

Having selected a material for armature stampings, the next phase of the design problem is the armature loading. This is measured in amp. conductors per in. of armature periphery. This loading can be increased by adding to the number of conductors in slots of the armature or by increasing the current density within those conductors. The first method will result in deepening of armature slots, with an undesirable increase in leakage flux. The second method is simpler if practicable. Current density is measured in amp. per sq. in., and a comparison with industrial practice is shown by Fig. 2(C). Higher values are being used in aviation practice. Such values are possible only when most effective cooling is utilized.

Current density will also affect the length of the commutator and the number of carbon brushes. It is common practice to use four, six or eight brushes on the larger size machines. As the number increases the length of the commutator, and therefore, the weight, decreases. Higher current densities in brushes will also decrease the length of the commutator. Aviation design practice has led the way into considerable higher densities. Where brush current densities of 40 to 100 amp. per sq. in. are common in industrial machines, aviation generators have brush densities of 180 to 230 amp. per sq. in. at normal rating. At the intermittent rating of generators and motors, these values can be increased 50 to 100 percent. It is known that densities as high as 600 amp. per sq. in. could be used if it was possible to place sufficient cooling on the brushes.

Values of armature loading that have been achieved by aircraft electrical designers are shown in Fig. 2(D). Since current densities are higher, this loading contrasts sharply with industrial practice.

A final measure of design practice is a comparison of output constants to eliminate the consideration of speed. A comparison of values is given in Fig. 2(E). The higher values of aviation practice result from use of increased loading and better steels.

Cooling Affects Weight

It is evident that cooling the machines has tremendous influence on the weight. Aviation is concerned with operation at the higher altitudes and in rarefied atmospheres. The reduced density of the air reduces the weight and the amount of cooling from a given volume of air. The lowering of ambient temperature with altitudes partially compensates for the reduction in density. But even with this lowering of ambient temperature, Fig. 3, the cooling effectiveness at 40,000 ft. is approximately 40 percent that at sea level.

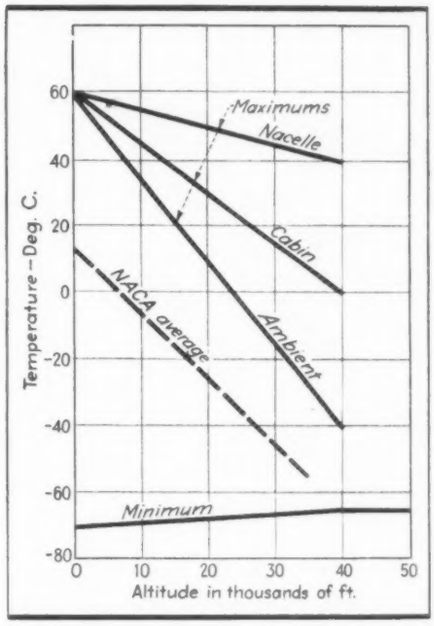


Fig. 3—Temperature changes encountered by aircraft at different altitudes.

Experience soon demonstrated that cooling by fans, whether of the radial or axial type was inadequate above 20,000 ft. Such fans deliver a given pressure differential, but as density decreases so will the differential and the cooling effectiveness. A fan-cooled generator is shown by Fig. 4(A). It was necessary to go to ram or blast cooling of generators early in the war. Such a design is shown by Fig. 4(B). The inlet tube opens forward or into a scoop. With this design the speed of the airplane counteracts the reduced density of the air. There is a slight drop in cooling effectiveness, but it is approximately 80 percent of the sea level values instead of 40 percent. The problem of cooling a motor or inverter in the cabin and without the advantage of blast cooling is difficult.

The trend in weight for other components

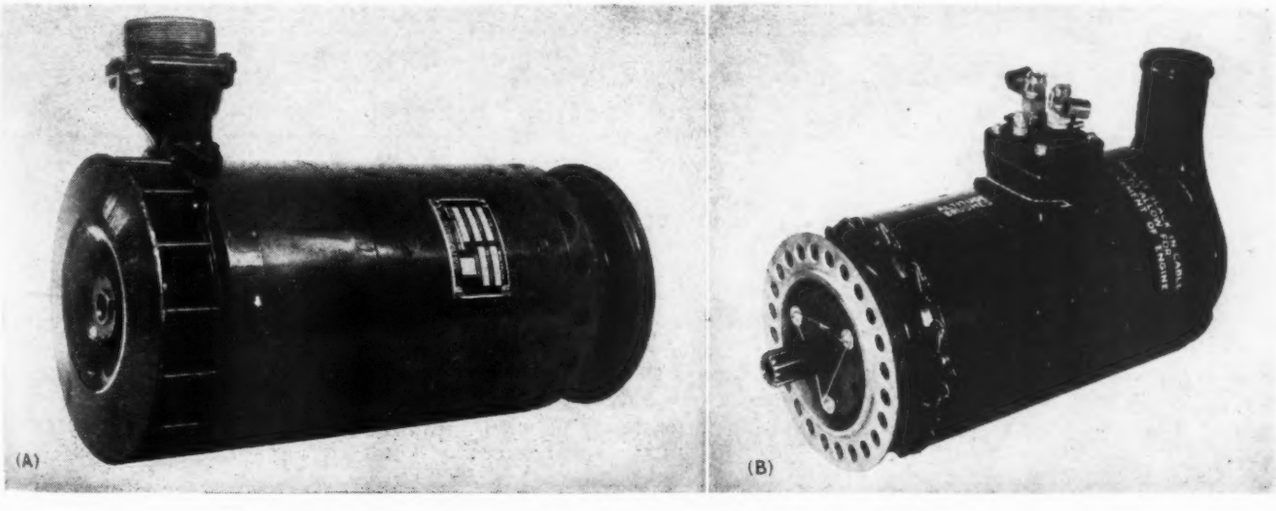
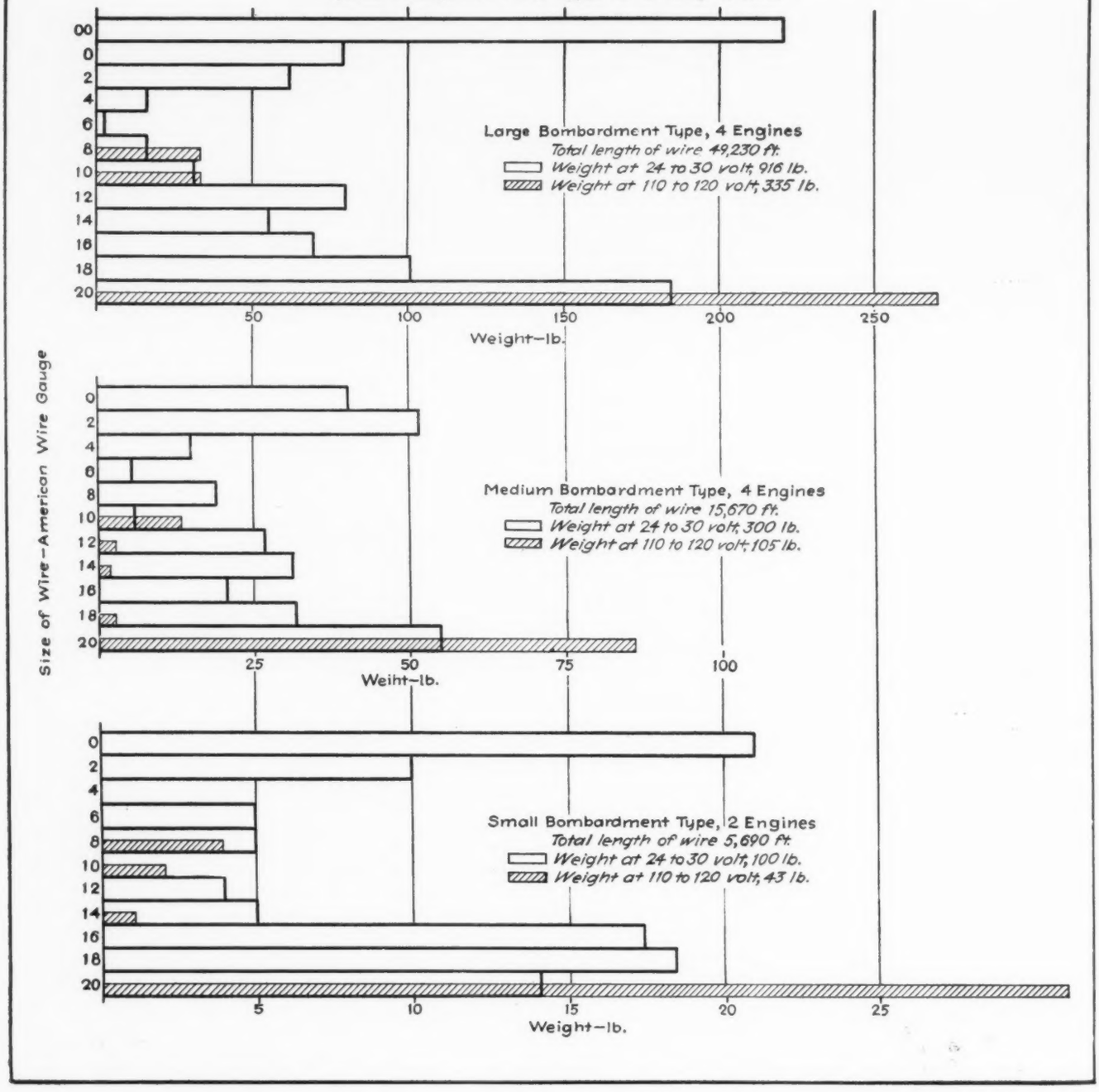


Fig. 4—(A) Fan cooled aircraft generator. (B) Blast cooled aircraft generator.

Table II—Distribution and Weight of Wiring of Different Sizes, At Two System Voltages in Three Types of Military Aircraft



of electrical systems can be summarized briefly. In wiring, there has been slight progress in the reduction of weight primarily as a result of the development of new types of insulation. Where current values of 100 amp. or more are used, aluminum conductors show the greatest possibilities of reduction in weight. The advent of the higher voltage system will do far more than any improvement in conductors to reduce the weight of wiring. For example, use of the 120 volt system would cut the weight of wiring to approximately 30 percent of its present value.

The change in wire size distribution in the three airplanes, in Table I, is shown by Table II. This chart is based on a change from a 24 to 30 volt d.c. to a 110 to 120 volt d.c. system. All of the theoretically possible reduction in weight is not realized because use of cable smaller than size AWG 20 is impractical.

Little progress has been made in the reduction of battery weights. There has been an appreciable trend towards the use of engine generator units as a substitute for batteries for two reasons. First, the weight of the engine generator unit is less, and second, they deliver power at regulated voltage as long as they receive fuel whereas batteries discharge in a short time, and deliver a steadily decreasing terminal voltage.

Progress has been made in reducing weight of lighting fixtures but the overall weight of lighting equipment tends to decrease but slightly because of the steady demand for natural and better lighting in the interior of airplanes.

It is hoped that the knowledge and experience gained by the aircraft electrical designer can be disseminated and utilized for better, cheaper and lightweight household and industrial appliances.

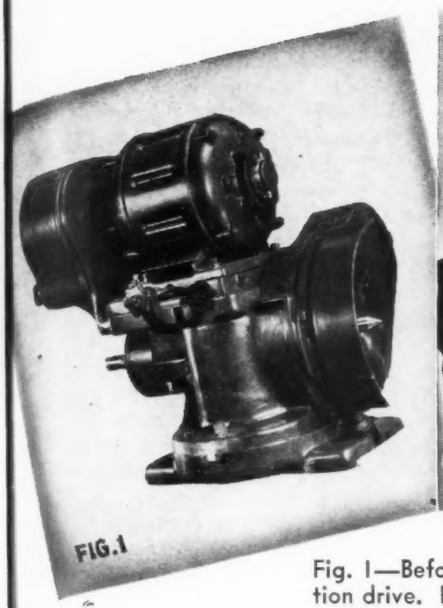


Fig. 1—Before redesign. Old type headstock with exposed motor and reduction drive. Fig. 2—After redesign. (A) New headstock with integrated motor and drive. (B) Close-up view with cover raise showing simplicity of design.

Simplified Variable-Voltage Drive Is Key to Improved Mechanical Design

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How development of a simplified constant-torque speed control system proved to be the key to obtaining many mechanical improvements. Step by step discussion of problems in redesigning typical headstock and their solutions. Series variable-voltage drive explained and its operating characteristics and limitations discussed.

IN REDESIGNING machines for the highly competitive peacetime economy, essential mechanical advances sometimes depend upon the development of new or improved electrical systems.

For example, the key to the realization of every pre-established mechanical design objective in simplifying a cylindrical grinding machine proved to be the successful development of a new electrical speed control system. The significance of the improvements stemming from the series variable-voltage drive can be best understood by discussing the design problems and their solution.

The precision and utility of a cylindrical grinding machine are determined by the headstock and the grinding wheel spindle assembly. Coordinated engineering resulted in a complete redesign of the headstock with many operating advantages. The old design is shown in Fig. 1 and the redesign in Fig. 2 shows the improvement.

Objectives Defined

From the beginning of the development program for an improved design of the headstock, three axioms were recognized:

1. A simple design is the best design,
2. Specific advantages accrue from close coordination of electrical and mechanical components,
3. Modern machines must not only consider, but favor, the operator.

In addition, definite and specific objectives included: (a) Extreme rigidity of the work spindle, (b) a wide and easily adjustable speed range, and (c) a harmonious structural form of pleasing appearance. Previous attempts to attain these objectives had met with only partial success as shown by Fig. 1; the greatest obstacle invariably being the electric motor and its associated speed reduction mechanism. Despite the many types of adjustable-speed fractional horsepower drives, for this particular project none quite met the specifications. Objectionable characteristics included: Size too big, first cost too high, operation too rough, or design too complicated. The successful culmination of the development program, therefore, depended on the procurement of a suitable electric drive.

Rotational Requirements

Reduced to its barest fundamentals, the rotational requirements of a headstock of this type are: (a) There shall be a number of

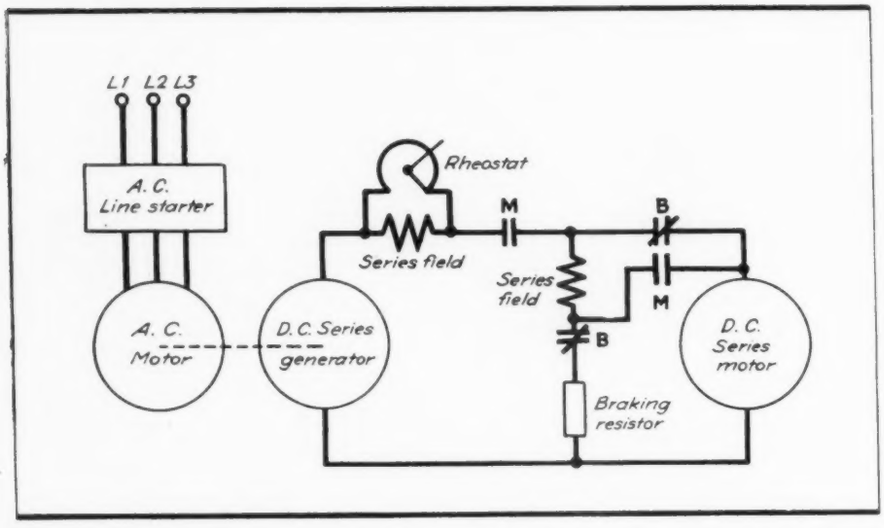


Fig. 3—Schematic diagram of series variable-voltage drive. Dynamic braking, obtained by reversing series motor field, insures quick stopping of motor.

adjustable speeds, and (b) the motor shall have speed-torque characteristics to permit theoretically constant rate of metal removal by the grinding wheel.

Constant horsepower motors with a base speed of 200 to 300 r.p.m. are physically too large and too costly. Speed range is normally limited to 4 to 1, although integral horsepower motors of 6 to 1 and even 8 to 1 range have been developed. Manufacturers of variable speed transmissions have tried valiantly to offset the inherent limitations of low speed constant horsepower motors by using mechanical reductions between a relatively high speed motor and a countershaft. These mechanical reduction devices include belt or chain systems, metallic variable ratio rings, and variable pitch pulleys. They can provide constant horsepower, constant torque, or combinations of both depending on the design and the motor characteristics.

From a mechanical point of view, few of these drives lend themselves readily to designs requiring acute conservation of space. In most instances, they do have the desirable feature of being self-contained. All reduction type drives require two or more sets of rotating components in addition to the motor. In machines such as grinders, this means that each rotating member is a potential source of vibration and therefore undesirable.

Constant Torque Drive Applied

Because of these limitations, a practical wide-range fractional horsepower constant torque drive was finally developed. The drive uses a substantially conventional d.c. series motor and a d.c. series generator, and is known as the series variable-voltage drive. As shown in Fig. 3, no exciter is required. It is not to be confused with the separately excited adjustable-voltage Ward Leonard system.

Because of the particular grinder design, only a single reduction was used with a driving motor of sufficient horsepower to adequately power the work spindle at low speed. The motor was rated 3/4 horsepower at 1,800 r.p.m. full field speed and operated over a 10 to 1 speed range from 180 to 1,800 r.p.m. Because of the duty cycle, a fractional horsepower motor frame sufficed.

With the motor meeting all desired qualifications, a headstock was designed to specifically match the electric drive. The innovations in mechanical design directly attributable to the application of the variable voltage motor are: (a) Low symmetrical form harmonizing with the whole machine, (b) single reduction V-belt work-spindle face-plate drive, and (c) maximum accessibility. Incident to these, is also a ball bearing mounted work spindle and a ball bearing face plate assembly of advanced design.

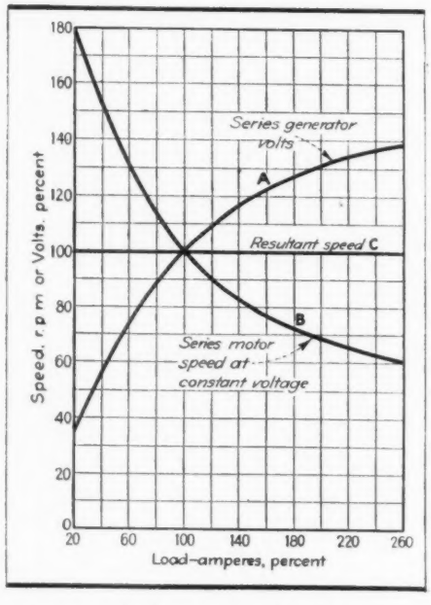


Fig. 4—Idealistic speed-load characteristic of a series motor connected in series with a series generator.

The development of a d.c. variable-voltage system for this headstock drive is a continuation of a trend toward d.c. equipment on machine tools. This has been especially noticeable in larger tools, such as planers and milling machines, where the increased cost of the d.c. equipment is justified by increased production and by the simplification of the mechanical arrangement.

The series variable-voltage system is ideal for driving headstocks and loads of a similar nature. A stepless speed range of 10 to 1 is available without the use of speed changing belts or gears. The speed is controlled by a small rheostat conveniently mounted adjacent to the pushbuttons used for starting and stopping the machine.

While the mounting arrangement of the drive for the headstock itself was ideal, the power supply unit for the motor also requires space. Conventional adjustable-voltage drives require a motor-generator set consisting of a d.c. generator, d.c. exciter, and a.c. driving motor. On small machines, such as a small grinder, the electrical equipment must be mounted on or in the machine to facilitate moving the machine and to conserve floor space. Therefore, the fewer units required for the power supply the better.

In this instance, since the hydraulic pump

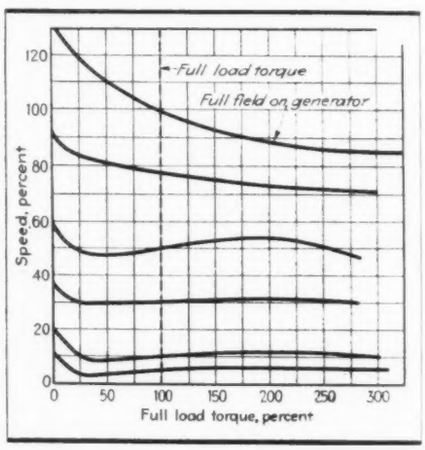


Fig. 5—Actual speed regulation curves obtained with the series variable-voltage drive are relatively flat.

for the grinder is driven by a constant-speed squirrel cage induction motor of relatively large horsepower as compared to the headstock motor, the small self-excited series generator is merely belted to it.

Operating Characteristics

Theoretically, the series variable-voltage system shown schematically in Fig. 3 produces the idealistic characteristic speed load curve shown in Fig. 4. The series generator voltage load characteristic is shown by curve A, whereas the series motor characteristic is

the inverse curve B. When the two machines are operated in series, the resultant voltage characteristic is the sum of the two or curve C. Therefore, the low voltage of the series generator at light load limits the inherently high no-load speed of the series motor; and the high voltage of the series generator at heavy loads increases the speed of the series motor at these heavy loads.

However, under actual operating conditions typical speed-torque curves are obtained such as shown in Fig. 5. The top curve shows the speed torque characteristics with motor and generator both operating at full field. By shunting the generator field, the family of speed torque curves shown is obtained. Those for low speeds are practically flat, except for a rather abrupt increase in speed at light loads. The speed regulation of the drive, based on the difference between actual full-load and no-load speeds, is relatively high. However, for all practical purposes on most drives, speed regulation is good because there is sufficient friction in the load to preclude an actual increase in speed at no load.

The series variable-voltage drive has the characteristics of high starting torque inherent in a series motor. With the rheostat set for 1/10 maximum speed, the drive can exert a starting torque of 3 to 4 times full-load torque. Therefore, it is not necessary to apply an oversize motor just to obtain the required starting torque.

The simplicity of the control scheme is shown by Fig. 3. The a.c. motor and a.c. linestarter are those for driving the hydraulic pump motor. The rheostat shunting the series field provides a means of controlling the speed. Starting and stopping the drive is done by closing the two contacts *M*, which connect the motor to the generator.

With this arrangement, the motor starts quickly when the contacts *M* are closed and accelerates up to a speed as determined by the setting of the rheostat. Once the proper speed has been determined, the rheostat need not be changed when starting and stopping.

When the stop button is depressed, dynamic braking insures quick stopping. Pushing the button opens contacts *M* and closes contacts *B*. This in effect reverses the series field of the motor and connects it and the motor armature in series with a braking resistor. The motor then becomes a series generator and dynamic braking brings it to a quick stop. In practice the contacts *B* are simply back contacts on the contact *M* so that only one magnetic contactor with two normally open contacts and two normally closed contacts are used.

With the drive simplified to this extent, the amount of space required for electrical components compares favorably to any type of mechanical drive that could be worked out and requires less space than any of the other known types of wide speed range electrical drives.

When all factors are considered, the series variable-voltage system is economical and compares favorably in cost, to other types of drives. For headstock applications, it is smaller and is considerably less expensive than any electronic drive available at the present time. While the drive does not have the great range in speed of the electronic and some of the other drives, it has all that is required for applications of this type, and therefore, a wider speed range is of no advantage.

Case Histories In Patent Law

H. A. TOULMIN, JR.

[Editor's Note: These explanations of the law in specific patent cases are not intended to supplant the services of a patent attorney, whose accurate advice is necessary to handle individual patent problems. The purpose of these explanations is to give a better general understanding of patent law.]

NO. 3

QUESTION: Can a subcombination patent be legitimately used as a means of preventing appropriation of parts of a machine by others when the inventor also owns a patent on the machine as a whole?

ANSWER: The Supreme Court of the United States has recently spoken on this important question. A patent was granted on a combination embodied in an inventor's complete machine, without allowance of subcombination claims, and therefore would not prevent free use of the subcombination. The inventor feared that someone would take a portion of his main machine covered by his main patent and that subcombination so taken would be the heart of his machine which his patent did not cover.

He therefore applied for another patent on the subcombination. The Patent Office refused to grant it. The Supreme Court reversed the Patent Office saying that the denial of a patent on the subcombination would deprive the inventor of the benefit of the exclusive right to use a subcombination in the ways specified by the patent laws. If this were not the case, it would leave the public free to use and to thus appropriate a part, however important, of the inventor's complete machine, even though the whole machine were patented.

The Court pointed out that it was a settled practice of the Patent Office to allow subcombination as well as combination claims and that this practice should be continued. See Special Equipment Company v. Coe, 324 U. S. 370.

The Court further held that the Federal Statutes permit, and it is a settled practice of the Patent Office, many times sustained by the Supreme Court, to allow claims to the broad combination and also to subcombinations of parts of the machine. It is evident that the value of the main patent would be greatly impaired if a subcombination invention could be freely used by others. This the Supreme Court prevented by its decision.

NO. 4

QUESTION: Does the failure of the patentee to make use of a patented invention affect the validity of the patent?

ANSWER: The Supreme Court of the United States has recently again held in the negative. A patentee may rightly use his patent as a protection against misappropriation of his invention, even though it is not used. Continental Paper Bag Company v. Eastern Paper Bag Company, 210 U. S. 405; Crown Die & Tool Company v. Nye Tool Works, 261 U. S. 24, 34; Woodbridge v. United States, 263 U. S. 50, 55; Fox Film Corp. v. Doyal, 286 U. S. 123, 127; Hartford-Empire Co. v. United States, 323 U. S. 386, 433.

The patentee in this case intended use of the subcombination patent to prevent others from appropriating it and by that means prevent others from appropriating an essential part of a complete machine. This was in no way inconsistent with the patentee making other permissible uses of the subcombination patent. The Court went so far as to say that the execution of the patentee's declared purpose to prevent appropriation of either of his inventions, whether used separately or together, would not prevent his licensing others to make, use and vend the subcombination, on terms which would adequately protect the monopoly of both of his inventions to which he was entitled by the patent laws. He could license one or both of his patents covering the full combination or the subcombination as he pleased. Special Equipment Co. v. Coe, 324 U. S. 370.

FINE PITCH GEARS

Inspection and Tolerances—IV

American Gear Manufacturers Association Standard 236.01, June 1946, for inspection and tolerances of gears of 20 diametral pitch and finer. For the preceding sections, 1 to 6 inclusive, see Product Engineering for September, October and November 1945.

SECTION VII—GEAR BLANKS FOR FINE PITCH GEARS

SCOPE

141. The following recommendations for gear blanks are intended chiefly as a guide for the manufacture of gears which will come within the quality specifications given in Section I. They are not to be regarded as inflexible rules but rather as the minimum average requirements of gear blank quality which is compatible to the finished gear, as found in the experience of many manufacturers of fine pitch gears. The principles of this standard can be suited to individual requirements without penalizing the general idea of standardization.

BLANK ACCURACY

142. The accuracy to which gears can be produced is largely affected by the design of the gear blank. Unless the gear blank has the required accuracy and rigidity it penalizes the production of good gears. When a gear is designed with a hole, the hole should be large enough to adequately support the blank during the machining of the gear teeth and yet not so large as to cause distortion. Face widths should be wide enough, in proportion to the outside diameters, to avoid springing and to permit obtaining flatness in cardinal surfaces. Very short bore lengths should be avoided wherever possible. It is possible, however, to machine relatively thin blanks in stacks provided the surfaces are flat and parallel with each other. When this technique is employed the whole stack of blanks is considered as a single blank rather than individual components. Quite frequently gear blanks must be designed with hubs. Attention should be given to wall sections of the hubs. Too thin a section will not permit proper clamping of the blank during the machining operation and may also affect proper mounting of the gear. Another fairly common practice is that of producing pinions, and on occasion, gears, integral with their shafts. This practice lends itself to increased accuracy if the proper precautions are taken. Among these precautions is that of avoiding deflection of the shaft by having the shaft length and shaft diameter well proportioned to the gear or pinion diameter.

The following are the most common faults in gear blank designs, which have proved troublesome:

(a) Very small bores compared to the outside diameter
(b) Very large bores compared to the outside diameter
(c) Narrow face widths with large outside diameter
(d) Short bore length
(e) Thin hub walls
(f) Gears integral with shafts that lack stiffness

GEAR BLANK TERMS

143. **Mounting surface** is that portion of the gear blank which positions the gear along its axis in the final assembly. It is designated on the drawing by the abbreviation "M. S.".

144. **Registering surface** is that portion of the gear blank which is used for mounting the gear in the machining operation. It must be made to the same relative accuracy as the mounting surface. It is designated on the drawing by the abbreviation "R. S.".

145. **Outside diameter** is the diameter of the addendum (outside) circle and will be larger in the blank, than in the finished gear, if an allowance for topping is made. In a bevel gear it is the diameter at the crown circle. In the throated worm gear it is the maximum diameter of the blank.

146. **Bore diameter** is the diameter of the hole concentric with the axis of rotation of the gear.

147. **Taper of bore** is the amount that the diameter of a nominally straight bore tapers in one inch of length, excluding bell mouth.

For example: If the diameters of a bore, measured at two points along the bore axis one-half inch apart are 0.500 in. and 0.5004 in., the bore taper is 0.0008 in. per inch of length.

148. **Bell mouth of bore** is an extreme taper at the ends of the bore and is not related to taper. The presence of bell mouth is equivalent to a shortening of the bore, therefore no hard and fast rule can be established for this condition. In general the total portion of the bore which is bell mouthed should not exceed 25 percent of the entire bore length.

149. **Runout of outside diameter** is the total indicator reading obtained by rotating the gear through 360 deg. about its axis.

In order to avoid interference in the teeth of fine-pitch gears which have not been topped, the runout of the outside diameter of the blank is held proportionately closer for the finer than for the coarser pitches. Conditions may occur where the maximum runout of the outside diameter may exist on a blank made to the nominal diameter. If a close backlash tolerance were specified in such a gear interference might develop when the high teeth come together. For this reason it is recommended that care be exercised to reduce the outside diameter to the low limit when the blanks are made to the maximum runout limit. (See Table I).

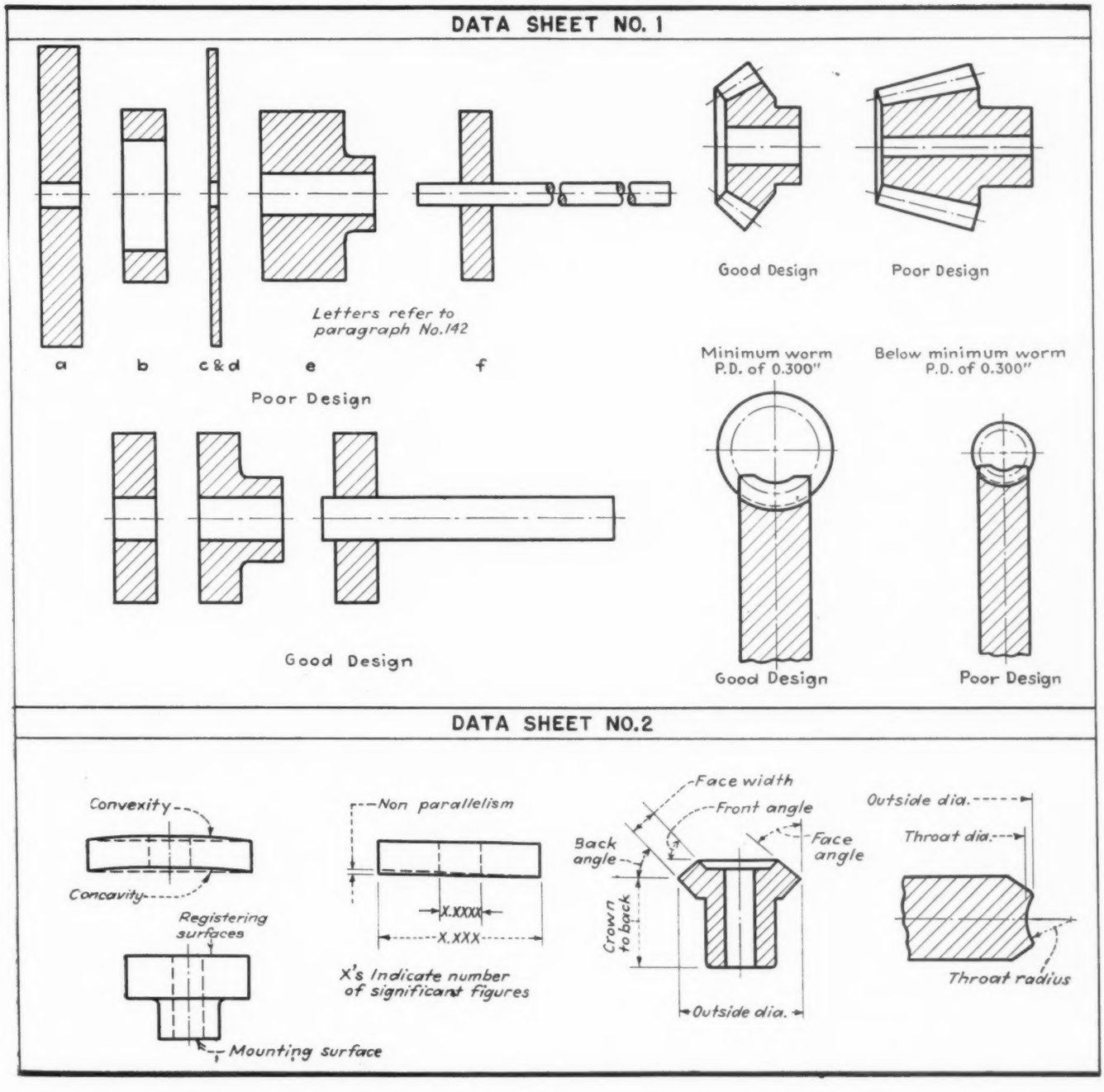
Table I—Runout of Outside Diameter With Bore

Diametral Pitch	Class	Runout Indicator Reading
20 to 40	Commercial 1, 2 and 3	0.0030
40 to 80	Commercial 1, 2 and 3	0.0020
80 and finer	Commercial 1, 2 and 3	0.0010
20 to 40	Precision 1, 2 and 3	0.0020
40 to 80	Precision 1, 2 and 3	0.0015
80 and finer	Precision 1, 2 and 3	0.0010

These tolerances are to be considered binding on a manufacturer or seller only when specifically agreed upon in writing.

150. **Finish of bore** is the quality of the surface finish of the bore and is covered in Section X—Surface Finish.

151. **Concavity** of mounting and registering surfaces is the amount these surfaces deviate from a plane to a concave surface. The deviation from a true plane shall be measured along a radial line by an indicator or straight edge. (See Data Sheet No. 2).



152. **Convexity** of mounting and registering surfaces is the amount these surfaces deviate from a plane to a convex surface. This measurement is taken by the same method as described for concavity. (See Data Sheet No. 2).

This condition is not desirable since the "mounting and registering surfaces" of the gear must fit corresponding flat surfaces in assembly or in machining the teeth. If the surfaces are convex the gear will not be adequately supported near the teeth. This causes the blank to spring during the machining of the teeth resulting in poor finish and inaccurate gears.

153. **Lateral runout** of mounting and registering surfaces is the total reading obtained by placing an indicator against these surfaces of the gear blank and rotating it through 360 deg. The lateral runout is referred to a one inch radius.

For example: If an indicator is placed at a radius of 2 in. from the bore axis, and reading of 0.001 in. is obtained, the lateral runout is 0.0005 in. per inch of radius. It is desirable to make this check as near to the pitch circle of the gear as possible.

154. **Non-parallelism** of mounting and registering surfaces is the maximum rate in inches, per inch of radius, at which the planes of the mounting and registering surfaces approach each other. Such non-parallelism is particularly harmful in gear blanks which are machined in stacks.

DATA SHEETS

155. Data Sheet No. 1 is intended as a guide for the designer and shows a few examples of well porportioned gear blanks. Data Sheet No. 2 shows the elements of various types of gear blanks, their nomenclature and method of specifying the tolerances.

GENERAL REMARKS

156. Tolerances, where given, are total deviations from nominal size. This has been done to permit shop preference for the bi-lateral or uni-lateral methods of tolerance dimensioning.

157. The Data Sheets, Nos. 1 and 2, illustrate typical cases and

are intended for use only as guides. Judgment must be exercised in applying the information properly.

158. This standard is intended for fine pitch gears. The general principles can be applied to blanks for coarser pitch gears.

Table II—Classes and Tolerances

	Commercial Class			Precision Class		
	1	2	3	1	2	3
Diameter of Bore	0.002	0.001	0.0007	0.0005	0.0002	0.0002
Taper of Bore*	0.001 per in. of length Max. 0.002	0.0007 per in. of length Max. 0.001	0.0005 per in. of length Max. 0.0007	0.0003 per in. of length Max. 0.0005	0.0002 per in. of length Max. 0.0002	
Concavity of Mounting and Registering Surfaces	0.001 per inch of radius for rigid blanks. 0.0005 per inch of radius for flexible blanks. Total not to exceed 0.003			0.0005 per inch of radius for rigid blanks. 0.0003 per inch of radius for flexible blanks. Total not to exceed 0.0015		
Convexity of Mounting and Registering Surfaces	None			None		
Lateral Runout	0.002 per in. of radius. Max. 0.004	0.0015 per in. of radius. Max. 0.0025	0.001 per in. of radius. Max. 0.002	0.0007 per in. of radius. Max. 0.0015	0.0005 per in. of radius. Max. 0.001	
Parallelism	0.002 per in. of radius. Max. 0.004	0.0015 per in. of radius. Max. 0.0025	0.001 per in. of radius. Max. 0.002	0.0007 per in. of radius. Max. 0.0015	0.0005 per in. of radius. Max. 0.001	

* No portion of the taper of the bore must exceed bore dia. limit.
Note: The outside diameter tolerance of spur and helical gears should be obtained from Standard No. 207.01 — 20 Degree Involute Fine Pitch System for Spur Gears.
The tolerance for face angle, back angle, crown to back tolerance and outside diameter of bevel gears should be obtained from Standard No. 206.01 — Fine Pitch Straight Bevel Gears.

These tolerances are to be considered binding on a manufacturer or seller only when specifically agreed upon in writing.

SECTION VIII—PIN MEASUREMENTS OF FINE-PITCH INVOLUTE SPUR GEARS

INTRODUCTION

159. While the quality classification of fine-pitch gears is predicated on the composite check, as described in Section I, the use of pins for determining tooth thickness, or more specifically, the space width, is a useful adjunct. The pin measurement is especially useful for checking tooth thickness while the gear is still on the arbor during the machining operation prior to the final check with a master gear.

It should be understood that the results obtained by a pin measurement will not reflect tooth profile errors. For this reason, the method should be considered a supplement to the composite check rather than a substitute.

SCOPE

160. Tables are provided for involute spur gears made in accordance with A.G.M.A. 20 deg. Involute Fine-Pitch System for Spur Gears, No. 207.01. For general explanations of pin measurement of gears see A.G.M.A. Reference Sheet for Involute Calculations. No. 131.01.

The tables at 1 diametral pitch are as follows:

Table I—Standard Gears, External
1.5708 in. circular thickness, for gears having 18 and more teeth.

Table IA—Standard Long Addendum Pinions
Circular thickness in accordance with values in Table II of above Standard (207.01) for pinions having 10 through 17 teeth.

Table II—Standard Gears, Internal
1.5708 in. circular thickness, for gears having 32 and more teeth. For internal gears having fewer than 32 teeth special tooth proportions will be required. Individual calculations must be made for such cases.

TABULATION

161. Pin Diameter: The pin diameter is 1.680 in. for standard gears and 1.920 in. for standard long addendum pinions. The principal reason for adopting the 1.680 in. diameter is that pins of this size are satisfactory for both external and internal gears. Refer to Table III for pin diameters for various diametral pitches.

Racks:
The pin measurement dimensions for standard racks are shown in Fig. 1, page 135.

Thickness Factor:
Small changes of thickness can be calculated satisfactorily by using a thickness factor, k_m. The tables include this factor.

$$k_m = \frac{\text{Change in circular thickness}}{\text{Change in pin measurement}}$$

Pressure Angle at Center of Pin: Values of ϕ_m are listed in the tables.

EXAMPLES OF USE OF TABLES

EXAMPLE 1

Number of Teeth	N	30 (external)
Pressure Angle	ϕ	20 deg.
Diametral Pitch	P	24
Backlash Allowance	Δt	0.002 in.
From Tables, for 1 P:		
Measuring Pin Diameter	d	1.680 in.
Measurement Over Pins	M	32.239 in.
Thickness Factor	k_m	0.411
For 24 P:		
Measuring Pin Diameter	d	0.070 in.
Measurement Over Pins	M	1.3433 in.
Standard Circular Thickness	t	0.06545 in.

To obtain the desired 0.002 in. backlash allowance, the pin measurement must be decreased as follows:

$$\Delta M = \frac{\Delta t}{k_m} = \frac{0.002}{0.411} = 0.0049 \text{ in.}$$

Final pin measurement = 1.3433 — 0.0049 = 1.3384 in.

EXAMPLE 2

Number of Teeth	N	80 (internal)
Pressure Angle	ϕ	20 deg.
Diametral Pitch	P	48
Backlash Allowance	Δt	0.001 in.
From Tables, for 1 P:		
Measuring Pin Diameter	d	1.680 in.
Measurement Between Pins	M	77.705 in.
Thickness Factor	k_m	0.342
For 48 P:		
Measuring Pin Diameter	d	0.035 in.
Measurement Between Pins	M	1.6188 in.
Standard Circular Thickness	t	0.0327 in.

To obtain the desired $\Delta t = 0.001$ in. backlash allowance, the pin measurement must be increased as follows:

$$\Delta M = \frac{\Delta t}{k_m} = \frac{0.001}{0.342} = 0.0029 \text{ in.}$$

Final pin measurement = 1.6188 + 0.0029 = 1.6217 in.

Table I—Pin Measurements of Standard Involute Spur Gears—External

1 Diametral Pitch 1.680-in. Pin Diameter Pressure Angle—20 deg.

No. of Teeth	Dimension Over Pins, Inches	Thickness Factor	At Center of Pin	No. of Teeth	Dimension Over Pins, Inches	Thickness Factor	At Center of Pin	No. of Teeth	Dimension Over Pins, Inches	Thickness Factor	At Center of Pin	No. of Teeth	Dimension Over Pins, Inches	Thickness Factor	At Center of Pin
N	D_M	k_m	ϕ_m-Deg.	N	D_M	k_m	ϕ_m-Deg.	N	D_M	k_m	ϕ_m-Deg.	N	D_M	k_m	ϕ_m-Deg.
10	For Standard Long-Addendum Pinions see Table 1A			61	63.236	0.389	21.425	111	113.253	0.378	20.810	161	163.261	0.374	20.565
11				62	64.257	0.389	21.404	112	114.265	0.378	20.803	162	164.268	0.374	20.562
12				63	65.237	0.389	21.383	113	115.254	0.378	20.796	163	165.261	0.374	20.558
13				64	66.257	0.388	21.363	114	116.265	0.378	20.789	164	166.268	0.374	20.555
14				65	67.238	0.388	21.343	115	117.254	0.378	20.783	165	167.261	0.374	20.552
15															
16				66	68.258	0.387	21.342	116	118.265	0.377	20.776	166	168.269	0.374	20.548
17				67	69.239	0.387	21.306	117	119.255	0.377	20.770	167	169.261	0.374	20.545
18	20.220	0.435	24.065	68	70.258	0.386	21.288	118	120.265	0.377	20.763	168	170.269	0.373	20.542
19	21.156	0.431	23.990	69	71.240	0.386	21.270	119	121.255	0.377	20.757	169	171.261	0.373	20.539
20	22.225	0.430	23.827	70	72.259	0.386	21.253	120	122.265	0.377	20.751	170	172.269	0.373	20.536
21	23.167	0.426	23.675	71	73.241	0.385	21.237	121	123.255	0.377	20.745	171	173.261	0.373	20.533
22	24.229	0.425	23.534	72	74.259	0.385	21.221	122	124.266	0.377	20.739	172	174.269	0.373	20.530
23	25.175	0.421	23.403	73	75.242	0.385	21.205	123	125.256	0.377	20.733	173	175.262	0.373	20.527
24	26.232	0.420	23.282	74	76.259	0.385	21.189	124	126.266	0.377	20.727	174	176.269	0.373	20.524
25	27.183	0.418	23.169	75	77.243	0.384	21.174	125	127.256	0.377	20.722	175	177.262	0.373	20.521
26	28.235	0.417	23.064	76	78.260	0.384	21.160	126	128.266	0.376	20.716	176	178.269	0.373	20.518
27	29.189	0.414	22.966	77	79.244	0.384	21.146	127	129.256	0.376	20.711	177	179.262	0.373	20.515
28	30.237	0.414	22.874	78	80.260	0.384	21.132	128	130.266	0.376	20.705	178	180.269	0.373	20.512
29	31.195	0.412	22.788	79	81.244	0.383	21.118	129	131.257	0.376	20.700	179	181.262	0.373	20.509
30	32.239	0.411	22.706	80	82.261	0.383	21.105	130	132.266	0.376	20.695	180	182.269	0.373	20.507
31	33.200	0.409	22.629	81	83.245	0.383	21.092	131	133.257	0.376	20.690	181	183.262	0.373	20.504
32	34.241	0.408	22.557	82	84.261	0.383	21.079	132	134.266	0.376	20.685	182	184.269	0.373	20.501
33	35.204	0.407	22.489	83	85.246	0.382	21.067	133	135.257	0.376	20.680	183	185.263	0.373	20.499
34	36.244	0.406	22.424	84	86.261	0.382	21.055	134	136.267	0.376	20.675	184	186.269	0.373	20.496
35	37.208	0.404	22.362	85	87.247	0.382	21.043	135	137.258	0.376	20.670	185	187.263	0.373	20.494
36	38.245	0.404	22.304	86	88.262	0.382	21.032	136	138.267	0.376	20.665	186	188.269	0.373	20.491
37	39.212	0.403	22.249	87	89.247	0.382	21.021	137	139.258	0.375	20.660	187	189.263	0.373	20.488
38	40.246	0.402	22.196	88	90.262	0.382	21.010	138	140.267	0.375	20.656	188	190.270	0.372	20.486
39	41.215	0.401	22.145	89	91.248	0.381	20.999	139	141.258	0.375	20.651	189	191.263	0.372	20.483
40	42.247	0.400	22.097	90	92.262	0.381	20.989	140	142.267	0.375	20.647	190	192.270	0.372	20.481
41	43.217	0.399	22.051	91	93.249	0.381	20.978	141	143.258	0.375	20.642	191	193.263	0.372	20.478
42	44.249	0.399	22.007	92	94.262	0.381	20.968	142	144.267	0.375	20.638	192	194.270	0.372	20.476
43	45.220	0.398	21.964	93	95.249	0.381	20.958	143	145.259	0.375	20.633	193	195.263	0.372	20.473
44	46.250	0.397	21.924	94	96.263	0.380	20.948	144	146.267	0.375	20.629	194	196.270	0.372	20.471
45	47.222	0.396	21.885	95	97.250	0.380	20.939	145	147.259	0.375	20.625	195	197.263	0.372	20.469
46	48.251	0.396	21.848	96	98.263	0.380	20.930	146	148.267	0.375	20.621	196	198.270	0.372	20.466
47	49.225	0.395	21.812	97	99.250	0.380	20.921	147	149.259	0.375	20.617	197	199.263	0.372	20.464
48	50.252	0.395	21.777	98	100.263	0.380	20.912	148	150.267	0.375	20.613	198	200.270	0.372	20.462
49	51.227	0.394	21.743	99	101.251	0.380	20.903	149	151.257	0.375	20.609	199	201.263	0.372	20.460
50	52.253	0.394	21.712	100	102.264	0.380	20.894	150	152.268	0.375	20.605	200	202.270	0.372	20.458
51	53.228	0.393	21.681	101	103.251	0.380	20.886	151	153.259	0.375	20.601	201	203.264	0.372	20.456
52	54.253	0.393	21.651	102	104.264	0.379	20.878	152	154.268	0.374	20.597	300	302.272	0.369	20.308
53	55.230	0.393	21.622	103	105.252	0.379	20.870	153	155.260	0.374	20.593	301	303.268	0.369	20.307
54	56.254	0.392	21.594	104	106.264	0.379	20.862	154	156.268	0.374	20.589	400	402.273	0.368	20.232
55	57.232	0.392	21.568	105	107.252	0.379	20.854	155	157.260	0.374	20.586	401	403.270	0.368	20.232
56	58.255	0.391	21.542	106	108.264	0.379	20.846	156	158.268	0.374	20.582	500	502.274	0.367	20.186
57	59.233	0.391	21.517	107	109.253	0.379	20.839	157	159.260	0.374	20.579	501	503.271	0.367	20.186
58	60.255	0.390	21.493	108	110.264	0.378	20.831	158	160.268	0.374	20.575				
59	61.235	0.390	21.470	109	111.253	0.378	20.824	159	161.260	0.374	20.572				
60	62.256	0.389	21.447	110	112.265	0.378	20.817	160	162.268	0.374	20.569				

Table IA—Pin Measurements of Standard Long-Addendum Pinions—External

With Diameters and Tooth Thickness According to American Gear Manufacturers Association Standard For Spur Gear Tooth Form (Adopted 1935) 1 Diametral Pitch 1.920-in. Pin Diameter Pressure Angle—20 deg.

No. of Teeth	Outside Diameter Inches	Circular Thickness, Inches	Dimension Over Pins, Inches	Thickness Factor	At Center Of Pin	No. of Teeth	Outside Diameter Inches	Circular Thickness, Inches	Dimension Over Pins, Inches	Thickness Factor	At Center Of Pin
N	D_o	t	D_M	k_m	ϕ_m-Deg.	N	D_o	t	D_M	k_m	ϕ_m-Deg.
10	12.830	1.873	13.441	0.615	35.349	14	16.362	1.703	17.223	0.544	30.721
11	13.713	1.830	14.268	0.596	34.044	15	17.245	1.660	18.067	0.528	29.757
12	14.596	1.788	15.343	0.577	32.850	16	18.128	1.617	19.085	0.513	28 847
13	15.479	1.745	16.181	0.560	31.749	17	19.011	1.575	19.932	0.499	27.983

Table II—Pin Measurements of Standard Involute Spur Gears—Internal

1 Diametral Pitch 1.680-in. Pin Diameter Pressure Angle—20 deg.

No. of Teeth	Dimension Between Pins, Inches	Thickness Factor	At Center of Pin	No. of Teeth	Dimension Between Pins, Inches	Thickness Factor	At Center of Pin	No. of Teeth	Dimension Between Pins, Inches	Thickness Factor	At Center of Pin	No. of Teeth	Dimension Between Pins, Inches	Thickness Factor	At Center of Pin
N	D_M	k_m	ϕ_m-Deg.	N	D_M	k_m	ϕ_m-Deg.	N	D_M	k_m	ϕ_m-Deg.	N	D_M	k_m	ϕ_m-Deg.
30	27.665	0.296	16.162												
31	28.628	0.298	16.281	76	73.704	0.341	18.674	121	118.702	0.350	19.189	166	163.715	0.354	19.420
32	29.670	0.301	16.426	77	74.689	0.341	18.693	122	119.712	0.350	19.196	167	164.708	0.354	19.423
33	30.635	0.303	16.561	78	75.705	0.341	18.711	123	120.702	0.350	19.203	168	165.715	0.354	19.427
34	31.674	0.306	16.686	79	76.690	0.342	18.728	124	121.712	0.350	19.210	169	166.708	0.354	19.430
35	32.641	0.307	16.801	80	77.705	0.342	18.744	125	122.703	0.350	19.217	170	167.715	0.354	19.433
36	33.677	0.310	16.908	81	78.691	0.342	18.760	126	123.713	0.350	19.223	171	168.708	0.354	19.436
37	34.646	0.311	17.008	82	79.706	0.343	18.776	127	124.703	0.351	19.229	172	169.715	0.354	19.439
38	35.680	0.313	17.102	83	80.691	0.343	18.792	128	125.713	0.351	19.235	173	170.709	0.354	19.442
39	36.651	0.314	17.191	84	81.706	0.343	18.808	129	126.703	0.351	19.241	174	171.716	0.354	19.445
40	37.683	0.316	17.274	85	82.692	0.343	18.823	130	127.713	0.351	19.247	175	172.709	0.354	19.448
41	38.655	0.317	17.353	86	83.707	0.344	18.837	131	128.704	0.351	19.253	176	173.716	0.354	19.451
42	39.686	0.319	17.426	87	84.693	0.344	18.851	132	129.713	0.351	19.259	177	174.709	0.355	19.454
43	40.658	0.320	17.496	88	85.707	0.344	18.865	133	130.704	0.351	19.265	178	175.716	0.355	19.457
44	41.688	0.321	17.561	89	86.693	0.344	18.879	134	131.713	0.351	19.271	179	176.709	0.355	19.460
45	42.661	0.322	17.623	90	87.707	0.345	18.892	135	132.704	0.351	19.277	180	177.716	0.355	19.463
46	43.689	0.323	17.682	91	88.694	0.345	18.905	136	133.714	0.351	19.283	181	178.709	0.355	19.466
47	44.664	0.324	17.738	92	89.708	0.345	18.917	137	134.704	0.351	19.289	182	179.716	0.355	19.469
48	45.691	0.325	17.792	93	90.695	0.345	18.929	138	135.714	0.352	19.294	183	180.709	0.355	19.472
49	46.667	0.326	17.844	94	91.708	0.345	18.941	139	136.705	0.352	19.299	184	181.716	0.355	19.475
50	47.692	0.327	17.893	95	92.696	0.346	18.953	140	137.714	0.352	19.304	185	182.710	0.355	19.477
51	48.669	0.328	17.939	96	93.708	0.346	18.965	141	138.705	0.352	19.309	186	183.716	0.355	19.480
52	49.694	0.329	17.983	97	94.696	0.346	18.976	142	139.714	0.352	19.314	187	184.710	0.355	19.483
53	50.672	0.329	18.025	98	95.708	0.346	18.987	143	140.705	0.352	19.319	188	185.716	0.355	19.486
54	51.695	0.330	18.066	99	96.697	0.346	18.998	144	141.714	0.352	19.324	189	186.710	0.355	19.489
55	52.674	0.331	18.106	100	97.709	0.347	19.009	145	142.706	0.352	19.329	190	187.716	0.355	19.492
56	53.696	0.331	18.144	101	98.697	0.347	19.019	146	143.714	0.352	19.334	191	188.710	0.355	19.495
57	54.676	0.332	18.181	102	99.709	0.347	19.029	147	144.706	0.353	19.339	192	189.716	0.355	19.498
58	55.697	0.333	18.216	103	100.698	0.347	19.039	148	145.714	0.353	19.344	193	190.710	0.355	19.501
59	56.677	0.333	18.249	104	101.710	0.347	19.049	149	146.706	0.353	19.349	194	191.716	0.355	19.504
60	57.698	0.334	18.281	105	102.698	0.348	19.058	150	147.714	0.353	19.354	195	192.710	0.355	19.507
61	58.679	0.334	18.312	106	103.710	0.348	19.067	151	148.706	0.353	19.359	196	193.716	0.355	19.510
62	59.699	0.335	18.342	107	104.699	0.348	19.076	152	149.714	0.353	19.364	197	194.710	0.356	19.512
63	60.681	0.335	18.371	108	105.710	0.348	19.085	153	150.707	0.353	19.368	198	195.717	0.356	19.514
64	61.700	0.336	18.399	109	106,699	0.348	19.094	154	151.715	0.353	19.372	199	196.710	0.356	19.516
65	62.682	0.336	18.426	110	107.710	0.348	19.102	155	152.707	0.353	19.376	200	197.717	0.356	19.518
66	63.701	0.337	18.452	111	108.700	0.349	19.110	156	153.715	0.353	19.380	201	198.711	0.356	19.520
67	64.683	0.337	18.477	112	109.711	0.349	19.118	157	154.707	0.354	19.384	300	297.719	0.358	19.682
68	65 702	0.338	18.502	113	110.700	0.349	19.126	158	155.715	0.354	19.388	301	298.715	0.358	19.683
69	66.685	0.338	18.526	114	111.711	0.349	19.134	159	156.707	0.354	19.392	400	397.719	0.360	19.762
70	67.703	0.339	18.549	115	112.701	0.349	19.142	160	157.715	0.354	19.396	401	398.717	0.360	19.762
71	68.686	0.339	18.571	116	113.711	0.349	19.150	161	158.707	0.354	19.400	500	497.720	0.361	19.810
72	69.703	0.339	18.592	117	114.701	0.349	19.158	162	159.715	0.354	19.404	501	498.718	0.361	19.810
73	70.687	0.340	18.613	118	115.712	0.350	19.166	163	160.708	0.354	19.408				
74	71.704	0.340	18.634	119	116.701	0.350	19.174	164	161.715	0.354	19.412				
75	72.688	0.340	18.654	120	117.712	0.350	19.182	165	162.708	0.354	19.416				

Table III—Pin Diameters for Various Pitches

P	*For Standard Gears* $d = \frac{1.680 \text{ in.}}{P}$	*For Pinions* $d = \frac{1.920 \text{ in.}}{P}$
20	0.08400	0.09600
24	0.07000	0.08000
28	0.06000	0.06857
32	0 05250	0.06000
36	0.04667	0.05333
40	0.04200	0.04800
48	0.03500	0.04000
60	0.02800	0.03200
64	0.02625	0.03000
72	0 02333	0.02667
80	0.02100	0.02400
96	0.01750	0.02000
100	0.01680	0.01920

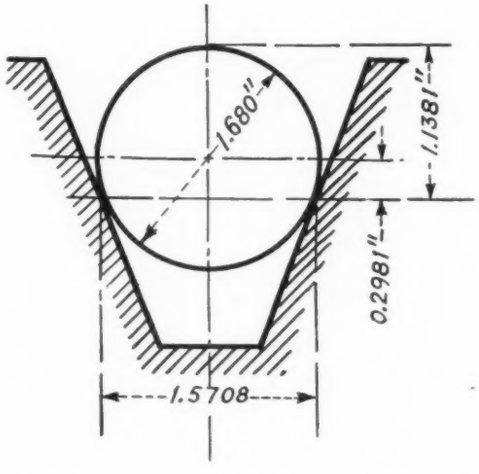


Fig. 1—Pin measurement of standard 20-deg. rack.

SECTION IX—FINE PITCH MASTER GEARS

SCOPE

162. This standard applies to 20 deg. pressure angle master gears of twenty diametral pitch and finer in the following general types:

Spur and helical gears
Worm and worm gears
Bevel gears; straight, spiral, Zerol, and hypoid gears.

DEFINITIONS

163. **A master gear** is a gear of known accuracy in every respect. It is usually made of wear resistant material and, by measurement, must be proved to be highly accurate in regard to bore size, lateral runout, tooth bearing, and total composite error as defined in Section I.

164. **A sample gear** is identical with the production gears and is retained for control or reference purposes. Such gears are usually made of the same material as the production gears and have the same cardinal dimensions.

SPUR AND HELICAL GEARS

165. **A standard master gear** for spur and helical gears is one in which the standard normal tooth thickness at the standard pitch diameter, or the corresponding pin or ball measurement, is within the following prescribed limits:

Diametral Pitch	Normal Tooth Thickness Tolerance from Standard. Inches	Approx. Corresponding Pin or Ball Measurement Tolerance from Standard. Inches
20 to 50	±0.0020	±0.0055
50 to 90	±0.0015	±0.0041
90 and finer	±0.0010	±0.0027

The outside diameter of the master gear should be corrected to the corresponding change in tooth thickness. The minimum whole tooth depth for standard master gears shall be $2.300/P + 0.003$ inches.

USE OF MASTER GEARS

166. Master gears are used primarily to check the total composite tooth error, tooth bearing and backlash. The measurement of these errors is obtained on a variable center-distance fixture as described in Section I. (See PRODUCT ENGINEERING, Sept. 1945, p. 698)

Fixed center-distance fixtures are not recommended because they do not record the tooth-to-tooth and total composite errors as the gear is rotated in intimate contact with the master.

167. A measurement of backlash is obtained with master gears by measuring the related reduction in center distance from a nominal setting. Lack of backlash is measured by a similarly related increase in center distance.

168. For fine-pitch gears, it is usually unnecessary to provide masters of special tooth thickness to check backlash. All of the active profile of gears which are made to backlash specifications given in Section V (see PRODUCT ENGINEERING, Oct. 1945, p. 712) and in accordance with Standard 207.01—20 Degree Involute Fine Pitch System For Spur Gears, can be checked with a standard master gear provided it is made with 50 percent greater tooth clearance than the clearance specified for the production gear.

169. Master gears are also used for charting errors of gear teeth on variable center distance devices equipped with recording mechanisms.

WORM AND WORM GEARS

170. Master worms are usually hardened and ground. Since the tooth form of worms is governed by the method of production due consideration must be given to the technique employed in producing the master worm.

171. Profile and lead error of fine-pitch worms can be checked by means of an optical comparator, see Section III. (See PRODUCT ENGINEERING, Oct. 1945, p. 710)

Runout and tooth thickness must be within the following limits:

Master Worm Class	Tooth Thickness Measured Over Wires Inches	Runout Total Indicator Reading Inches
1	±0.001	±0.001
2	±0.0005	±0.0005
3	±0.0003	±0.00025

172. Master worm gears are not hardened and ground but should be made of a wear resistant material. The material need not be the same as the production worm gear. The same equipment should be used to produce the master worm gear as that used for producing production gears. Tooth bearing and total composite error shall

be measured on a variable center-distance fixture with a master worm as described in Section III. The tolerance for master worm gears is as follows:

Class	Total Composite Error Inches
1	0.0013
2	0.0020
3	0.0010

BEVEL GEARS

Straight, Spiral, Zerol Bevel Gears and Hypoid Gears

173. Master straight bevel gears are not usually hardened and ground but may be made of a wear resisting material.

174. Master spiral, Zerol bevel and hypoid gears are hardened and ground in the coarser of the fine-pitch series. In the finer pitches they are usually made of a wear resistant material. In producing master gears of the bevel gear family it is important that a satisfactory tooth bearing be obtained. This is best accomplished by matching the bearing of the master to the developed bearing of production gears.

TOLERANCES

175. Master bevel gears are usually furnished in pairs. The total composite error in each gear of a pair shall be obtained as follows:

The gear being checked shall be placed in intimate contact with its mate on a suitable variable center-distance fixture which duplicates theoretical mounting distances and includes a movable head and an indicating device for measuring head displacements. See Section IV.

METHOD OF DETERMINING COMPOSITE ERROR IN SPUR, HELICAL AND BEVEL GEARS

176. The total composite error of a master gear is determined as follows:

A reading is taken of the master and its mating gear by rotating the master gear through 360 degrees. The master gear and work are then returned to their original positions. The master gear is then disengaged from the mate and turned on its axis a minimum of 90 degrees, establishing a new sequence of teeth. Care should be taken to mark a tooth and a space to show the original point of engagement. A minimum of four readings is taken. The average of these readings is the total composite error in the master gear.

177. In addition to the composite check a tooth bearing check is also required. See paragraph 88, Section IV.

Classes and Tolerances of Spur, Helical and Bevel Master Gears

Class	Tooth-to-Tooth Composite Error Inches	Total Composite Error Inches
1	0.0004	0.00100
2	0.0003	0.00050
3	0.0002	0.00025

178. A tooth bearing check is equally desirable for spur and helical master gears since it gives an indication of lead error. The desired bearing for spur and helical master gears shall be a minimum of 90 percent of the face width of the master without appreciable cross bearing.

Master Gear Blank Tolerances for Spur, Helical and Bevel Gears (all dimensions are in inches for all classes)

Hole size within*.....	+0.0001 −0.0000
Face runout of locating surfaces with respect to hole, total indicating reading per inch of diameter not to exceed...	0.0001
Runout of outside diameter, with respect to hole, total indicator reading.....	0.0010
Outside diameter size (with respect to adjusted diameter) 20 to 30 D. P.....	+0.000 −0.002
30 D. P. and finer.....	+0.000 −0.001

(Must be adjusted to correspond to the tooth thickness.)

* The hole size in master gears which must be left soft may require a more liberal tolerance, which will vary with each case.

MARKING FOR STANDARD SPUR AND HELICAL MASTER GEARS

The following is the minimum recommended marking for standard spur and helical master gears:

	Abbreviation	Symbol (Preferred)
Spur Gears		
Diametral Pitch	D.P.	P
Pressure Angle	P.A.	ϕ
Number of Teeth	xx T	N
Tooth Thickness at the standard pitch diameter*	T.Th. $= \frac{C.P.}{2} \pm$x.xxxx $= y.yyyy$	$t = \frac{p}{2} \pm$x.xxxx $= y.yyyy$
Standard Master Gear	Std. M.G.	Std. M.G.
Helical Gears		
Diametral Pitch (Transverse plane)	D.P. Trans.	P_t
Pressure Angle (Transverse plane)	P.A. Trans.	ϕ_t
Number of teeth	xx T	N
Helix Angle	H.A.	Ψ
Hand (of master)	R.H. L.H.	R.H. L.H.
Tooth thickness at the standard pitch diameter (transverse plane)*	T.Th. $= \frac{C.P.}{2} \pm$x.xxxx $= y.yyyy$	$t_t = \frac{p_t}{2} \pm$x.xxxx $= y.yyyy$
Standard Master Gear	Std. M.G.	Std. M.G.

* The tooth thickness at the standard pitch diameter is one-half of the standard circular pitch plus or minus the tolerance increment as determined by pin or ball measurement. See paragraph 165.

OPTIONAL MARKINGS

179. In addition to the above data pertinent information such as manufacturer, part number, etc., may be indicated.

SPECIAL MASTER GEARS

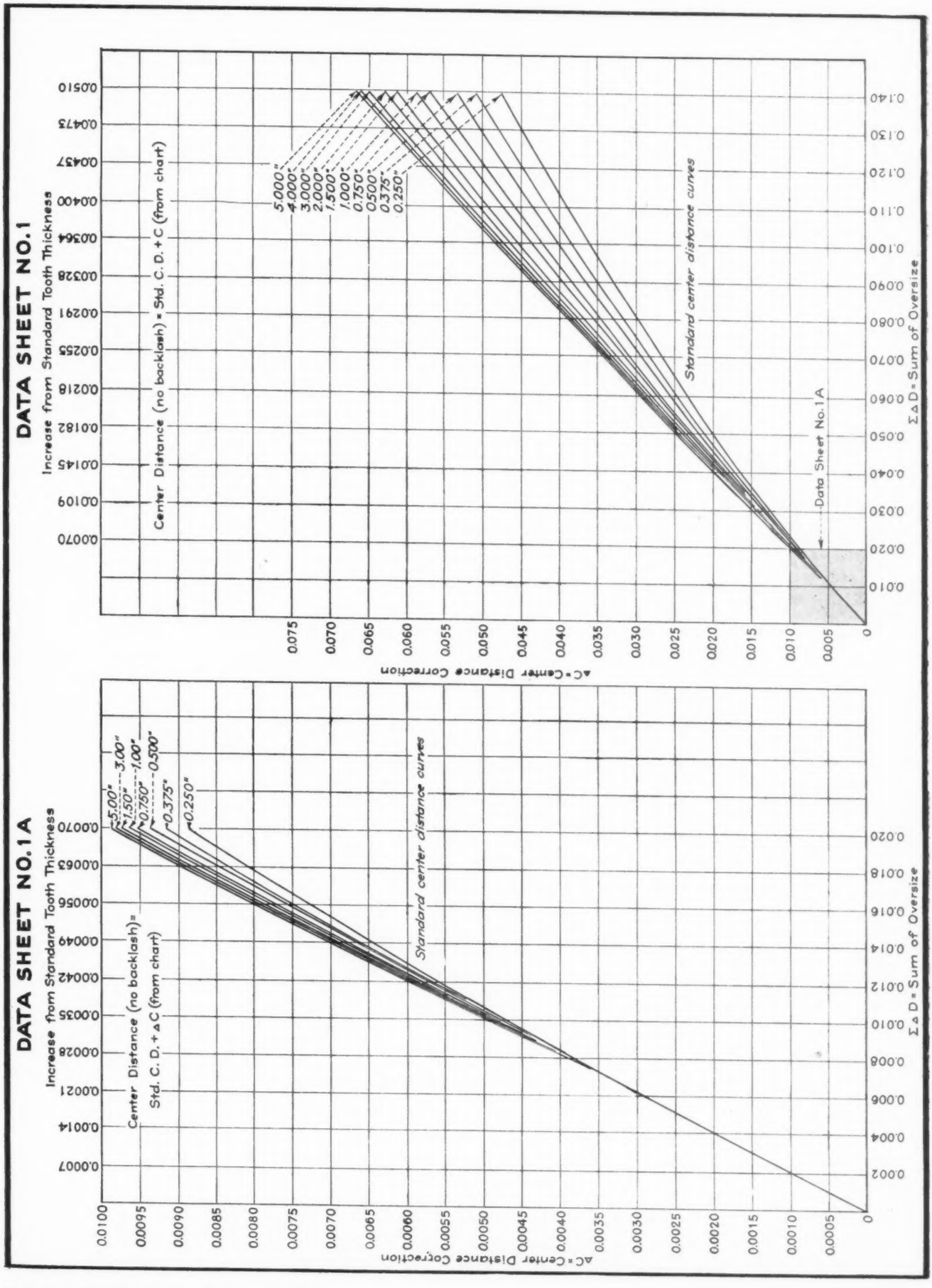
180. There may be fine pitch master gears which will deviate from this standard in regard to tooth proportions. Such master gears will require special marking as agreed upon by the user and manufacturer and should be identified as *Special*.

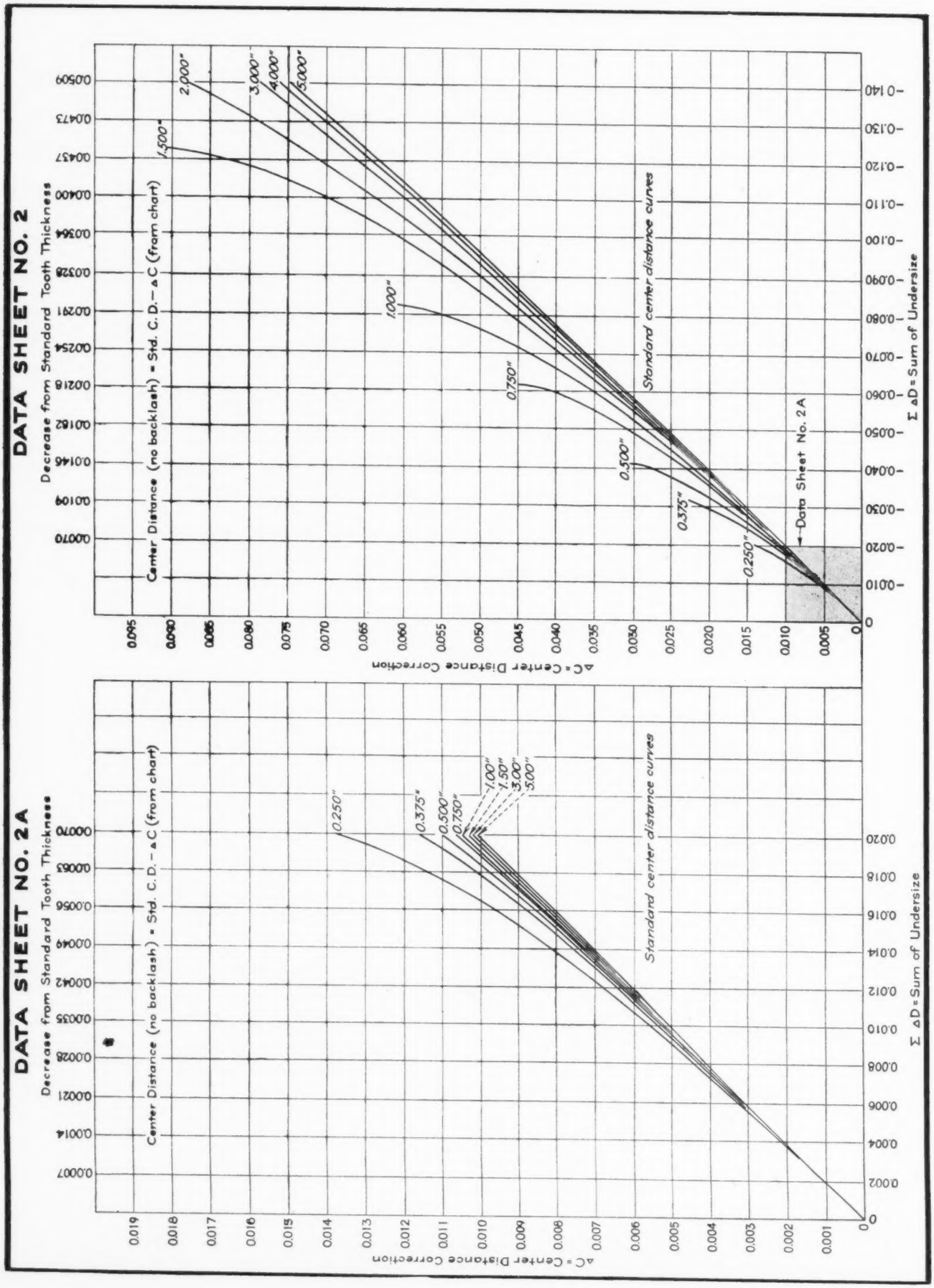
DATA SHEET

181. Data Sheets 1, 1A, 2, 2A are for determining the center distance correction for no backlash engagement of gears having nonstandard tooth thickness. Their derivation is based on the well known principle that a change in tooth thickness (on the standard pitch diameter) will result in a proportional change in center distance when meshing with a standard rack. This produces what is termed apparent diameter of the gear.

FORMULAS

1. $C = \frac{N + n}{2P}$
2. $\Delta\Sigma_t = \Sigma\Delta D \tan \phi$
3. $\Delta \text{ inv } \phi = \frac{\Delta\Sigma t}{2C}$
4. $\text{inv } \phi_1 = \text{inv } \phi \pm \Delta \text{ inv } \phi$
   $\Delta \text{inv } \phi$ is + for increased $\Sigma\Delta D$ and − for decreased $\Sigma\Delta D$
5. $K = \frac{\cos \phi}{\cos \phi_1}$
6. $C_1 = kC$
7. $\Delta C = C_1 - C$
8. D apparent $= D \pm \Delta D$


DATA SHEET NO.1
Increase from Standard Tooth Thickness
Center Distance (no backlash) = Std. C. D. + C (from chart)
5.000"
4.000"
3.000"
2.000"
1.500"
1.000"
0.750"
0.500"
0.375"
0.250"
Standard center distance curves
Data Sheet No.1A
ΔC = Center Distance Correction
ΣΔD = Sum of Oversize
DATA SHEET NO.1A
Increase from Standard Tooth Thickness
Center Distance (no backlash) = Std. C. D. + ΔC (from chart)
5.00"
3.00"
1.50"
1.00"
0.750"
0.500"
0.375"
0.250"
Standard center distance curves
ΔC = Center Distance Correction
ΣΔD = Sum of Oversize


DATA SHEET NO. 2
Decrease from Standard Tooth Thickness
Center Distance (no backlash) = Std. C.D. − ΔC (from chart)
ΔC = Center Distance Correction
Σ ΔD = Sum of Undersize
Standard center distance curves
Data Sheet No. 2A
1.500"
2.000"
3.000"
4.000"
5.000"
1.000"
0.750"
0.500"
0.375"
0.250"
DATA SHEET NO. 2A
Decrease from Standard Tooth Thickness
Center Distance (no backlash) = Std. C.D. − ΔC (from chart)
ΔC = Center Distance Correction
Σ ΔD = Sum of Undersize
Standard center distance curves
0.250"
0.375"
0.500"
0.750"
1.00"
1.50"
3.00"
5.00"

SYMBOLS:

P = Diametral Pitch
N = No. of Teeth in Gear
n = No. of Teeth in Pinion
ϕ = Std. pressure Angle
ϕ_1 = Pressure Angle at C_1
C = Standard Center Distance
C_1 = No backlash Center Distance
Δt = Increase or Decrease from Std. Tooth Thickness
ΔD = Increase or decrease from Std. Pitch Diameter
$\Sigma\Delta t$ = Sum of Increase or decrease from Std. Tooth Thickness
$\Sigma\Delta D$ = Sum of Oversize or Undersize
D = Std. Pitch Diameter

EXAMPLE NO. 1

Pinion	Gear
$N = 20$	$N = 60$
$P = 20$	$\Delta D = 0.005$ Oversize
$\phi = 20$ deg.	
$\Delta D = 0.045$ in. Oversize	

Sum of Oversize = $\Sigma\Delta D = 0.045 + 0.005 = 0.050$ in.

Std. Center Dist. = $C = \frac{20 + 60}{2 \times 20} = 2.000$ in.

Refer to the bottom abscissa of Data Sheet 1A and locate 0.050 in. on $\Sigma\Delta D$ scale. Continue vertically on this line until it intersects the 2.000 in. center distance curve, refer this point to the ordinate scale and read 0.0238 in. change in center distance.
Then: Center Distance (No Backlash)

$$C_1 = 2.000 + 0.0238 = 2.0238 \text{ in.}$$

EXAMPLE NO. 2

Pinion	Gear
$N = 40$	$N = 80$
$P = 80$	$\Delta t = 0.0015$ in. increase
$\phi = 20$ deg.	
$\Delta t = 0.0055$ in. increase	

$\Sigma\Delta t = 0.0055 + 0.0015 = 0.0070$ in.

$$C = \frac{40 + 80}{2 \times 80} = 0.750 \text{ in.}$$

Refer to the top abscissa of data sheet 1a and locate 0.0070 in. on the "Increase from Std. Tooth Thickness" scale. Continue vertically down this line until it intersects the 0.750 in. center distance curve, refer this point to the ordinate scale and read 0.00956 in. change in center distance.
Then: Center Distance (No Backlash)

$$C_1 = 0.750 + 0.00956 = 0.75956 \text{ in.}$$

EXAMPLE NO. 3

Pinion	Master Gear
$N = 40$	$N = 80$
$P = 80$	$\Delta D = 0.005$ undersize
$\phi = 20$ deg.	
$\Delta D = 0.015$ undersize	

Sum of Undersize = $\Sigma\Delta D = 0.015 + 0.005 = 0.020$ in.

$$C_1 = \frac{40 + 80}{2 \times 80} = 0.750 \text{ in.}$$

Refer to Data Sheet 2A at —0.020 and proceed as outlined above; read change in center distance equal to 0.01058 in.
Then: Center Distance (No Backlash) = $C_1 = 0.750 - 0.01058 =$ 0.73942 in.

182. Data Sheets 1A and 2A are enlargements of the section up to 0.020 in. sum of oversize and undersize shown on Charts 1 and 2.

183. Data Sheet No. 3 gives the recommended diameters, face widths, bore lengths and bore sizes of master gears.

184. Data Sheet No. 4 gives two suggestions of master gear design for use where extreme accuracy is required.

DATA SHEET NO. 3

Recommended Diameter and Hole Size of Master Gears

Diametral Pitch	Pitch Diameter Inches	Hole Size Inches*
20–29	3.0	1.0000
30–49	2.0	0.7500
50–79	1.5	0.5000
80 and finer	1.0	0.5000

* Note the hole sizes given above are subject to changes to suit the existing equipment either by the master gear manufacturer or user.

In no case should a master gear have fewer than 40 teeth

RECOMMENDED FACE WIDTH

Maximum face width = $\frac{D}{4}$ or $\frac{15}{P}$ whichever is smaller.

Minimum face width = $\frac{D}{12}$

The bore length should never be less than its diameter.

These tolerances are to be considered binding on a manufacturer or seller only when specifically agreed upon in writing.

DATA SHEET NO. 4

Two suggestions of Master Gear Design for Use Where Extreme Accuracy Is Required:

1.

Jarno Taper; 0.600 in. taper per foot
Bore size to suit gear diameter (see Data Sheet No. 3)
For use where gear rotates with spindle, such as on gear recorders.

2.

For use on centers

These designs permit greater accuracy by eliminating bore tolerance.

EXPLANATION OF ITEM NO. 165.

It can be shown both graphically and by means of calculations, that gears designed in accordance with AGMA Fine Pitch Standards 207.01* and 236.01*, can be inspected satisfactorily with master gears made in accordance with Section IX, Standard 236.01. These masters have been termed Standard Master Gears and check all of the active tooth profile of gears made within the backlash specifications of Table 8, Section V, Standard 236.01. This table covers a range from no measurable backlash to 0.006 in. backlash between two mating gears.

Although the most critical conditions exist at the transition points in Table 8, Section V, Standard 236.01, it can be shown by calculations that some clearance exists. A 70 pitch gear, having 0.0025 in. backlash meshing with a standard rack will have 0.0015 in. clearance when placed in intimate contact with each other. In order to increase the clearances to a practical point the use of a clearance factor of 2.300/P plus 0.003 in. for standard master gears is suggested. This will result in the clearance being increased by approximately 50 percent, or in this particular case will give a minimum clearance of 0.0039 inches.

* AGMA Standard 207.01—20 Degree Involute Fine Pitch System for Spur Gears.

* AGMA Standard 236.01—Inspection, Fine Pitch Gear Inspection and Tolerances.

INDUSTRY AND SOCIETIES

RESEARCH—STANDARDS—PATENTS

RAM JETS

Enthusiastic Navy revelation of the ram jet method of aircraft propulsion focuses attention on the simplicity and great power of this new engine, leading to speculation concerning its future possibilities, especially in comparison with other types of reaction motors. Made from the tail pipe of a P-47 airplane, shown below is the first working model of a ram jet engine ever flown.

A FLYING MODEL driven by an engine without moving parts at speeds greater than sound has been revealed by the Navy Department. One of its most closely guarded secret developments, the so-called ram jet engine has been successfully flown at speeds exceeding 1400 miles per hour.

Operating under special research contracts with the Bureau of Ordnance, the Applied Physics Laboratory of Johns Hopkins University, Silver Spring, Maryland, and 20 associated industrial organizations and universities have during the past year proved conclusively that the ram jet is a practical method of achieving high speed flight. The first successful flight was made June 13, 1945 at Island Beach, New Jersey.

The ram jet, developed for the Navy Bureau of Ordnance as a means of propulsion for guided missiles consists essentially of an open pipe. Oxygen is scooped into the front opening from the air during flight and compressed by the speed of the jet, fuel is injected and burned and the exhaust streams out the rear opening. The impulse thus produced by the escaping hot gases thrusts the jet through the air at a velocity of from 800 to 1500 miles per hour. This engine differs from other jet engines in that the air is "rammed" into the engine by the forward motion of flight rather than by a mechanical compressor.

The ram jet is frequently called the "flying stovepipe" because of its form and the fact that it has no moving parts. Actually, the first test jet was made from the exhaust pipe of a Thunderbolt plane. This modified exhaust produced more than the horsepower of the conventional reciprocating engine from which it was detached. Since they have no moving parts, no precision machine work is required and they can be produced rapidly and cheaply. The low cost feature is especially important for use in missiles that are expendable.

Description

The ram jet is composed of three principal sections; the forward part is known as the diffuser, the middle part is a combustion chamber, and the latter part is the exhaust nozzle. The arrangement of these parts is shown in the accompanying diagram. The function of the diffuser is to receive air at atmospheric pressure, and by expansion at subsonic speed increase its pressure to the highest value practicable. The combustion chamber furnishes a flame front that increases the momentum of the air, thus creating a back pressure supporting that developed in the diffuser. It is essentially from this phenomenon that the ram jet gets its thrust. The pressure of the air in the diffuser exerts a forward thrust on the diffuser inner walls, and the reaction is against the exhaust gases. Exhaust nozzles are either simple constrictors, straight ducts with no constriction, or a combination of constriction and expansion. The matching of the three sections of the ram jet involves complicated thermodynamic balancing.

How It Works

The compression furnished by a piston in an automobile engine is obtained in the ram jet with an inlet diffuser that converts the kinetic or velocity energy of the fast-moving air to pressure energy. In other words, the air moves into the engine with a high velocity. Once it is inside, the air is slowed down and compressed, creating a high pressure. That pressure built up can be used to discharge the air through the outlet at approximately the same velocity at which it entered the engine.

If the air is heated while moving slowly and under high pressure and discharged through a small opening, or nozzle, the outlet velocity can be made to exceed the inlet velocity by a wide margin. That increase in velocity at the outlet produces thrust or forward motion. It is reaction force similar to that experienced when shooting a shot gun.

A minimum speed, which is approximately sonic, is required before the ram jet will operate efficiently. (Catapults or auxiliary rockets are used to bring the ram up to required speed.) The reason for this is that shock waves, a phenomenon occurring at supersonic speed, play an important part in the propulsive action.

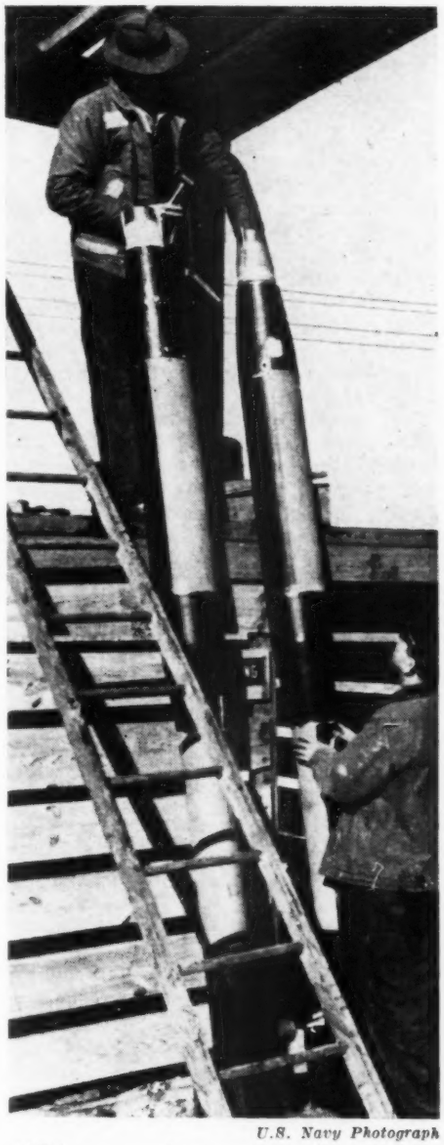
There are three types of diffusers that can be used to obtain desired recovery pressure. The simplest is a conical frustum with the small end used as the intake. Since the speed of the duct is supersonic relative to the intake air, and the speed of the air in the diffuser exit is subsonic and of the order of a few hundred feet per second, the air must somewhere pass through the sonic

speed while slowing down. The laws of conservation of energy, momentum, and mass flow require the formation of what is known as a normal shock wave for the change-over through the transonic region. Amazingly abrupt changes in pressure and velocity occur in this shock region, which is only a small fraction of an inch in thickness; for example, air moving at 1,800 ft. per sec. and at atmospheric pressure changes across this thin boundary to air at 900 ft. per sec. and nearly three times atmospheric pressure. Maximum thrust will be obtained when this normal shock wave is located at the nose or entrance of the diffuser.

The normal shock wave is characterized by a loss of entropy. This means that some of the energy of the gas has gone into heat irreversibly and the maximum pressure that can be recovered by the diffuser is somewhat reduced. Up to supersonic speeds less than twice the velocity of sound, this loss is not large. At the higher speeds, however, some means of approximating isenthropic compression in supersonic speeds is imperative. The simplest method is the use of the so-called reversed De Laval nozzle. In this arrangement the supersonic air stream is taken into a contracting duct and pressure is increased isentropically until sonic speed is reached at the minimum section, called the throat. From this point on, the duct expands as a normal diffuser, velocity subsonic and decreasing, and pressure increasing.

Another method of preventing large loss in entropy caused by compression at high supersonic speeds is to use a series of diagonal or conical shock waves such as are produced around the point of a cone, to obtain a multi-stage compression in the supersonic region of flow. The cone is a central body and the duct scoops the air from around the cone.

The combustion chamber of the ram jet is similar to that of other jet engines, with the added difficulty that stable flames must be established at considerably greater speed of air flow. A balance must be struck between a sufficiently elaborate igniter to burn the fuel completely and uniformly, while at the same time not producing an excessive drag force, thus reducing the available net propulsive force.



U.S. Navy Photograph

With no moving parts and openings at both ends, the ram jet model first flown by the Navy weighs only 70 lb. Workmen ready a model for flight.

Experimental Techniques

The basic difficulty in research work on ram jets is that they cannot be tested statically—either the ram jet must be moved through the air as in free flight or the air must be moved past the ram jet as in a wind tunnel.

There are several different ways of assaying the performance of ram jet engines. One can attach the intake of a ram jet directly to the source of air, and by appropriate adjustments of temperature and pressure reasonably simulate flight intake conditions. By making pressure surveys along the duct, thrust can be evaluated. There is also a method that more nearly simulates flight conditions by placing the ram jet intake in a supersonic air stream, the so-called free jet. Size considerations at the present time preclude the use of supersonic wind tunnels on flight models.

In free flight tests, both internal thrust and external drag are evaluated. The duct is first flown cold with a tail constriction, to simulate the flow conditions produced by a burner. Internal drag can be calculated from the flow condition and total drag is obtained from the deceleration rate shown from the velocity time curve. The unit is then flown with the burner operating, and the thrust of the unit calculated from the excess of thrust over drag.

In flight, it is necessary to employ telemetering methods. Pressures, accelerations, temperatures, fuel flows, and so forth, must all be converted into a radio signal that is broadcast from an antenna integral with the ram jet, and then picked up and analyzed by field equipment at the test site.

Other Reaction Engines

The thermal jet, or turbojet, engine will not be considered here because of the great amount of discussion they have received in the past several years. It should be remembered, however, that in this type of engine, about two-thirds of the energy of the jet is used to keep the cycle going, and only the remainder is available to provide thrust.

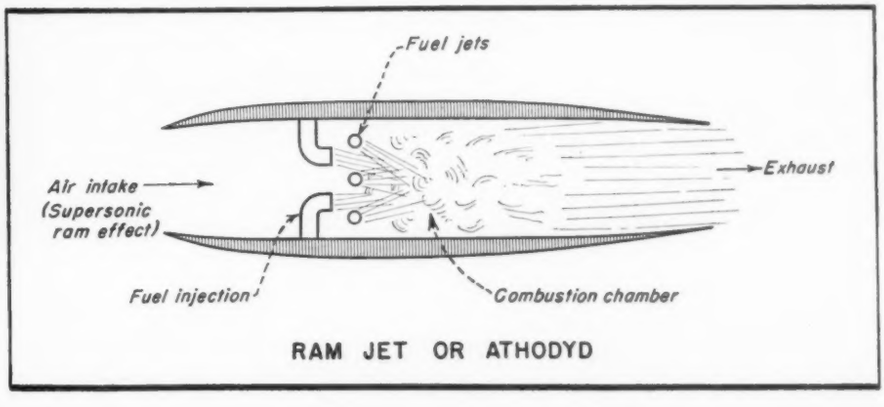
We can thus reduce the basic types of reaction engines to three: Rockets, pulse jets and ram jets or "athodyds."

True rockets have been discussed in detail in the "Rockets" story in the July "Industry and Societies" section. Several features of the liquid-fuel rocket motor should, however, be noted: Rockets operate independently of air. The liquid-fuel rocket requires the use of separate tanks for the propellants. The motor can thus be small and light. A motor capable of producing 100 lb. of thrust can be held in the palm of the hand. The German V2 rockets made use of a liquid fuel motor producing 60,000 lb. of thrust (30 tons), and this entire thrust was produced by a motor a little more than 6 ft. long and weighing about 1,000 pounds.

Pulse Jets

The second type of reaction engine is referred to by the British as an intermittent duct engine, and by American engineers as a pulse jet. It is also sometimes called a buzzbomb engine, because it was this kind of device that drove the German V-1 weapons, or buzzbombs.

In this engine the action must be started



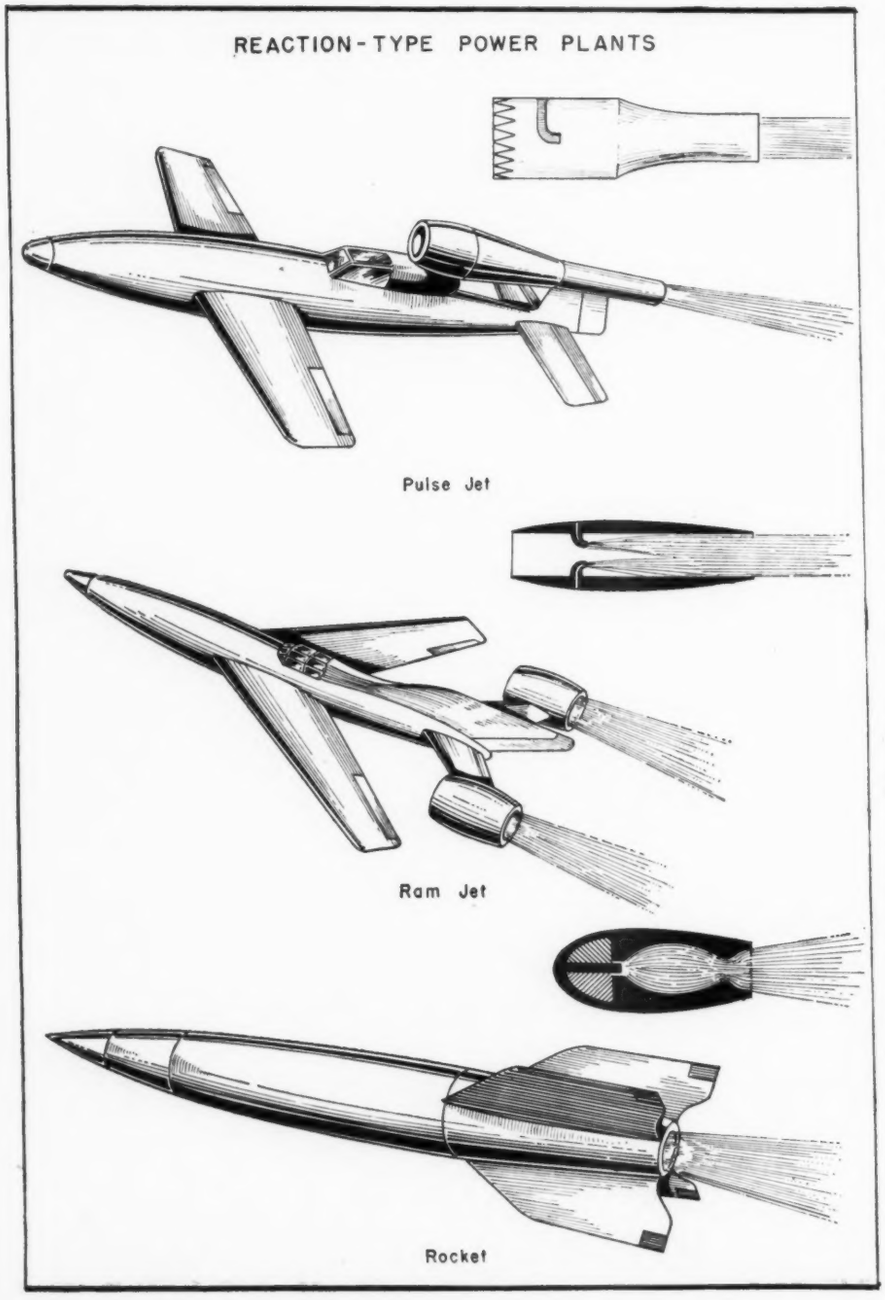
RAM JET OR ATHODYD

by compressed air or by rapid forward motion such as the Germans gave it with their buzzbomb launching ramps. When head pressure reaches a suitable level, the spring-controlled flap valves at the forward end of the tube are blown open and the air enters the combustion chamber, where it is mixed with gasoline and fired. The resulting blast blows the shutters closed. Expansion then forces the gases of combustion and the column of air out the rear, producing a jet. But this leaves an area of low pressure in the chamber, which automatically opens the shutters again. More air comes in, and the cycle is repeated.

The rate of the cycle depends on the resonance, and hence on the length of the tube. In the German buzzbomb engine the tube was about 11 ft. long, and the rate of firing about 40 times a second. It is because of this resonance effect that this device has been called the pulse jet.

The buzzbomb engine is poor as far as efficiency goes, but it is so light and simple, and so cheap to produce, it may nevertheless have many uses both in war and in peace. It was well suited to driving the German buzzbomb. The powerplant of the buzzbomb represented only 8 percent of the total starting weight of the craft, and permitted a payload that represented nearly half the launching weight.

Here is a brief comparison between a pulse jet and a ram jet of the same unit size functioning at 10,000 ft. at 400 miles per hour: The pulse jet is still delivering about 60 percent of its sea level thrust at a fuel consumption rate of approximately 4 lb. per hour per lb. of thrust. The ram jet is just about coming into its own, delivering about 20 percent of its net thrust, but with a fuel consumption of at least four times that of the pulse jet. If these same units are operated at sonic speed the drag on the pulse jet becomes so great that the effective net thrust is practically negative, the fuel consumption remaining constant, whereas, the ram jet is delivering about 60 percent of its net thrust, and now its fuel consumption in lb. per hour per lb. of thrust is about the same as that in the duct engine.



Ram Jet Performance

The third type of reaction engine is sometimes referred to as the continuous duct engine, though it is now becoming more familiar under the British name athodyd and the American ram jet. "Athodyd" is a classical-sounding title which has been made from the significant letters of the original British descriptive name, aero-thermal-dynamic-duct. Ram jet fits the engine better than any of the other names and is finding increasing favor.

Unlike rockets, ram jets require air for their operation (combustion). They do not operate efficiently at high altitudes.

Using the techniques described previously, it has been possible to demonstrate both in combustion tests in the laboratory and in free flight, the attainment of thrusts commensurate with those predicted by theory. Thrust alone, however, is not the most striking performance criterion. The complete ram jets weigh only 70 pounds, giving less than one-half ounce per horsepower as compared with one pound per horsepower for a conventional engine. Liquid-fuel rocket motors give about the same weight-horsepower ratio.

An ordinary aircraft engine driving a propeller burns half a pound of fuel per thrust horsepower hour.

The fuel rates and output for supersonic flight are enormous, but if the problem of attaining speeds in excess of 1,500 miles per hour at altitudes up to and beyond the maximum for present aircraft is to be overcome, the ram jet appears to be the most suitable engine.

At high speeds the chief competitor of the ram jet is the rocket. The ram jet requires about one-sixth as much fuel as the rocket at 2,300 miles per hour because it used oxygen from the atmosphere while the rocket carries its own. At extreme altitudes, ram jets are not effective because of lack of oxygen.

The possible application of the ram jet lies in three fields. Its greatest suitability is for supersonic aircraft, either piloted or pilotless. Because of its light weight and ease of fabrication, it also may be used for subsonic expendable missiles. The third possible application is for rotary wing aircraft where rotor wing tips speeds may approach the speeds desired for ram jet operation and where the economy of engine weight and freedom from power transmission problems compensate for a higher fuel economy.

RESEARCH NOTES

Aircraft Research in "Flying Laboratories"

Huge bombers, supplied by the Army Air Forces and stripped of their wartime gear, have been converted into flying laboratories by General Electric. Based at the company's newly completed Fight Test Center, which was officially opened with a national air research demonstration on June 21-22 at the Schenectady County Airport, the flying laboratories are operated for the development and testing of all types of gas turbine aircraft engines, radar and electronic aircraft equipment under actual flight conditions.

Jet propulsion testing is one of the main projects now under way at the Flight Test Center, which includes a huge hangar for the experimental aircraft, two floors of laboratories, workshops, engineering offices and a control tower for various types of radio equipment.

Presently based at the new center are two B-24 Liberators, a B-29 Superfortress and an autogyro. These flying laboratories, equipped with the newest scientific and engineering instruments, enable the G-E engineers to conduct tests on all types of aviation equipment under actual flight conditions, and thus facilitate the early perfection and production of material under development.

The Superfortress and one of the B-24's are now being used to test General Electric's powerful jet engines and to develop them for other types of aircraft. The other B-24 flying laboratory is used for the testing of the latest developments in the field of radar.

The torpedo-shaped jet, suspended from the bomb bay of the B-29, is operated while the bomber is in flight, instruments recording its operational characteristics. In the B-24, the jet engine is installed in the fuselage, which affords space for the testing of jet engines of varied sizes.

While many of tomorrow's airplanes with jet propulsion are still military secrets, engineers have made predictions of low-priced private planes of reasonably high economy, powered by gas turbine-driven propellers. They also predict helicopters with jet propulsion gases being emitted from the trailing tips of the propeller blades. Large commercial and passenger lines, G-E engineers point out, will be employing planes powered by gas-turbine propeller power plants. Commercial airlines already have expressed interest in obtaining gas turbines to power high-speed transport planes.

A TG-180 axial flow jet engine mounted in a retractile test cell in the bomb bay of a B-29 Flying Laboratory by the General Electric Company. The torpedo-shaped unit permits engineers to conduct tests under actual flight conditions.

Research Division of Army To Be on General Staff Level

Plans for the operation of a Research and Development Division on a level with the Personnel Intelligence, Training, Supply and Operations section of the General Staff were completed by General Eisenhower shortly before his departure for the Pacific. The Director of the new division has not yet been selected. Several ranking generals are under consideration. The Secretary of War has ordered outstanding general officers with scientific background to report to Washington for consultation prior to the selection of the director. In the meantime Colonel Gervais W. Trichel, of the War Department Special Staff New Developments Division, will serve as acting Director. The Research and Development Division will assume the functions of the New Development Division in addition to other duties.

The Division has primary interest in the application of national scientific resources to the solution of military problems. According to Dr. Edward L. Bowles, expert civilian consultant to the Secretary of War, who aided General Eisenhower in developing the plan for the new Division, the director will act as liaison between Army planning chiefs on one hand and industry and the universities and research laboratories on the other. Creation of the Division will result in the Army's coordination with existing civilian organizations in the development of new military techniques. Great civilian advances are foreseen as a result of much of the Army-sponsored research. Recognizing the abilities of civilian research group, the War Department will from time to time put before industrial and scientific experts broad problems which will be dealt with along lines which the civilian organizations have found to be most likely of productivity.

Summarizing his directive, General Eisenhower stated that "in order to ensure the full use of our national resources in case of emergency," the following general policies would be put into effect:

1. The Army must have civilian assistance in military planning as well as for the production of weapons.

2. Scientists and industrialists must be given the greatest possible freedom to carry out their research.

3. The possibility of utilizing some of our industrial and technological resources as organic parts of our military structure in time of emergency should be carefully examined.

4. Within the Army we must separate responsibility for research and development from the functions of procurement, purchase, storage and distribution.

5. Officers of all arms and services must become fully aware of the advantages which the Army can derive from the close integration of civilian talent with military plans and developments."

The Director of the Division will advise the Secretary of War and the Chief of Staff on all research and development matters, and will be responsible for the initiation, allocation, coordination and progress of research and development programs. He is also charged with demonstration of new or improved weapons, military equipment and techniques to the Army services which would use them. As a General Staff member, he will issue orders in the name of the Secretary of War and Chief of Staff on matters with which he is concerned.

All War Department activities dealing with research and development will come under this division. Civilian experts will be appointed in the near future to act as advisers to Colonel Trichel in setting up the activities of the Division. Information from this new Division will be distributed to you from this section of the Publications Branch of the Bureau of Public Relations.

Navy Uses Rockets To Probe Ionosphere

To EXPLORE A REGION of the atmosphere never before investigated, the Navy sent a miniature scientific laboratory 100 miles into the ionosphere on board a V-2 rocket fired at White Sands Proving Ground, Las Cruces, New Mexico on June 27. In the hope of obtaining answers to some of the problems about the universe Navy scientists at the

U. S. Navy Photograph

Naval Research Laboratory, Bellevue, D. C., have prepared the miniature laboratory which will be carried in the rocket's "warhead." Scientists stationed at many observation posts will operate a massive array of equipment to track the rocket in its six minute flight and record the data that it radios back to earth. The June 27 project was conducted in collaboration with Army Ordnance Engineers who have charge of the launching and flight of the missile. (See story on rockets in July "Industry and Societies" section). Photograph shows Navy scientists adjusting instruments in nose of V-2 preparatory to flight.

Closed Impression Die Forgings Research Program Announced

The Technical Committee of the Drop Forging Association has inaugurated a broad program of technical research under the direction of Professor John M. Lessells, of Lessells and Associates, Boston, Mass.

Studies of mechanical and metallurgical properties of closed impression die forgings will include tensile and hardness, torsion, impact, fatigue and damping properties. In view of the fact that approximately 80 percent of all service failures can be attributed to fatigue, considerable emphasis will be placed on this aspect of the investigations. Special facilities capable of testing specimens up to 3 inches will be available.

Studies of mechanical and metallurgical properties of other engineering materials will include the basic properties for closed impression die forgings and will be obtained on such materials as pearlitic malleable iron, cast steel, and alloy cast iron. Such materials as Meehanite, Proferal, Z Metal, and ArMaSteel will be included.

Compilation of comparative data obtained on properties of forged and cast materials will show how other engineering materials behave under stress as compared with the behavior of forged materials. Factual data on the comparative resistance to shock and repeated loads is of particular importance to designers.

Special investigations to be undertaken include:

1. Relative forgeability of materials used by the drop forging industry and the size of equipment required for a particular forging;
2. Effect of surface conditions on the fatigue strength, including decarburization, scale, seams and laps;
3. The importance of flow lines;
4. Comparison of soundness in metal products;
5. Effect of size on the fatigue strength.

Marine Engineering Research

Waterman Steamship Corp., Mobile, Alabama, has announced the establishment of a research project in the Engineering Research Division, Southern Research Institute. The program will include all phases of design and improvement of marine equipment and will be under the direction of E. N. Kemler, Head of the Division, and G. B. Clark of the Waterman Research Laboratory.

INTERNATIONAL STANDARDS ORGANIZATIONS HOLD PARIS MEETINGS

PROBLEMS OF PEACETIME ORGANIZATION will take the limelight at meetings being held in Paris during the early part of July by the executive bodies of three international standards organizations—United Nations Standards Coordinating Committee, International Standards Association, which was put "on ice" during the war because of Axis membership, and International Electrochemical Commission.

The Executive Committee of UNSCC met on July 10 to prepare an agenda and make preliminary arrangements for a plenary session, which will be held in London beginning on October 14, with the British Institute of Standards acting as host. The UNSCC group will at the same time hold a joint meeting in Paris with the Council of ISA (called together by the US and Denmark) to give effect to a resolution at the New York meeting of UNSCC last year recommending that ISA should be dissolved.

The United States will be represented on UNSCC's Executive Committee by Dr. Paul G. Agnew, vice-president and secretary of the American Standards Association, Great Britain by Mr. P. Good, director of the British Institute of Standards; Canada by Mr. G. Morrow, chairman of the Canadian Standards Association and chairman of the Executive Committee; and France by General Salmon, government representative on the French standards organization. The Soviet Union, which has now officially joined UNSCC, China and Brazil will be represented. These seven nations form UNSCC's Executive Committee. The Council of ISA consists of representatives from Denmark, Great Britain, Holland, Italy, United States and Russia.

Among the projects suggested for the UNSCC's Executive Committee are standardization programs in textiles, metal food containers, rayon, heat treatment of steel plastics terminology, airfield lighting, sheet and wire gauges, machine tools, automobiles, buildings, documentation and boiler construction codes.

UNSCC's membership is limited to United Nations representation. The 18 nations at present within the fold are: Australia, Belgium, Brazil, Canada, Chile, China, Czechoslovakia, Denmark, France, Great Britain, Holland, Mexico, New Zealand, Norway, Poland, South Africa, United States and Russia.

The future of UNSCC will be decided at the October meeting in London, when several outside nations, including, Switzerland, Sweden, Argentine and perhaps Spain, are expected to be present. The chances are that a new organization will be formed with the title of International Standards of Coordinating Association, which from the first would include countries outside the United Nations.

The International Standards Coordinating Association will have for its task the coordination of standardization in all fields of industry, but it is expected to confine its activities to coordination and not issue specific international standards as was the practice of ISA. The work of preparing standard specifications will continue to be dealt with by the standardization organizations in the different countries.

ISCA's guiding principle will have to be "standardization by consent," but it will undertake preliminary work and coordinating development based on economic need, not on academic theories of standardization. Obviously, success can be achieved only if there is free and full co-operation from the individual standardization bodies of the nations represented.

It will also be necessary to have the co-operation of other international organizations, such as the International Electrochemical Commission, which has 40 years procedural experience in the coordination of electrical standards. Chief purpose of IEC's Council meeting in Paris is to consider affiliation with ISCA.

It is expected that the work undertaken recently by another international group—the American-British-Canadian conferences on standardization—will continue on its present basis for the time being. A full report of the last meeting of this group, which concerned itself mainly with screw-thread standardization, appeared in the "Industry and Societies" section for April. It is reported that the next meeting of the United States-Canada-British group will be held in London, probably concurrently with the UNSCC meeting in October.

WASHINGTON NOTES

Many Congressional Issues Still Undecided

At this writing, Congress is in the home stretch of this session, with controversial bills of national moment no nearer to passage or settlement than when they were first introduced. Meanwhile, Congress was talking of adjourning until next year, rather than recessing until the fall, as was done last year.

Pieces of important legislation passed in this session are conspicuous by their paucity. We will attempt here, therefore, to list what Congress has left undone.

Federal Science Agency—This legislation is setting something of a record for long-term hedging, even in Congress, where indecisiveness is an art. Bills for federal support of research have been considered for almost two years. Readers of this column have been kept informed of the Kilgore-Magnuson research bills for the past year, culminating in the great "compromise" bill, S. 1850. (See April "Washington Notes" column.) Just when it appeared that some sort of federally sponsored research organization would become law, a new measure, the Mills bill, was introduced into the House. This represented the desires of the responsible scientists who had favored the old Magnuson bill before the "compromise". The Magnuson bill had the National Research Foundation administered by a board of scientists, with none of the tight federal control of the Kilgore bill, and retained in the Kilgore-Magnuson compromise.

Passed in the Senate—Early in July, the Kilgore-Magnuson National Science Foundation passed the Senate today by a vote of 46 to 18. All but one of the amendments urged by the senators whose proposed substitute measure failed to be adopted, were defeated separately. The coalition group succeeded only in preventing the inclusion within the Foundation of a Division of Social Sciences.

The first amendment, moved by Senator Smith of New Jersey to place control of the foundation in the hands of a nine-man board of experts instead of the single administrator, as provided in the Committee bill, failed on a tie vote, 34 to 34, and later on a motion to reconsider, by a vote of 35 to 34. The second, urged by Senator Chapman Revercomb of West Virginia, and Thomas C. Hart of Connecticut to strike out the restrictive patent provisions of the bill was lost. A further motion by Senator Hart of Connecticut to eliminate the proposed provision of undergraduate scholarships, and fix awards at graduate and fellowship levels, was defeated.

Elimination of the proposed social science division, also moved by Senator Hart, was adopted, the only one of the substitute proposals to survive.

The only limitation on the powers of the one-man foundation head achieved by opponents of the proposed organizational set-up was through an additional amendment, moved by Senator Raymond E. Willis, Republican of Indiana and accepted by Senators Harley M. Kilgore and Warren G. Magnuson, providing that the administrator's appointments of a deputy administrator, and of directors for the seven (not to exceed ten), divisions within the foundation, all at $12,000 a year, must be with the advice and consent of the Senate.

The salary of the administrator is fixed at $15,000. The compensation of the members of the board, and of the members of the divisional scientific committees to be appointed by the administrator with the advice and approval of the board is to be at the rate of $50 per day and expenses for each day spent by them on the business of the foundation.

The Board, and each divisional scientific committee is permitted to elect its own chairman from its own members and devise its own rules of procedure. The board is also permitted to appoint and prescribe the duties of an Executive Secretary of its own selection who shall receive compensation at a rate not exceeding $12,000 a year.

The functions of each of the divisions, however, shall be prescribed by the administrator, after receiving the advice of the board, including the seven presently specified, and "such additional divisions not to exceed three in number, as the administrator may from time to time establish after receiving the advice of the Board."

The Divisions presently specified in the bill are: a Division of Mathematical and Physical Sciences; of Biological Sciences; of Health and Medical Sciences; of National Defense; of Engineering and Technology; of Scientific Personnel and Education; and of Publications and Information.

In support of the proposed amendments it was argued by all the proponents that, while the administrator was instructed to consult the members of the board, he was free to accept or decline their advice, in all cases, which he might in discretion completely ignore.

In urging elimination of the social science division, which prevailed, and revision of the sections of the bill governing allocation of funds, which was lost on a voice vote, Senators White and Hart, both of whom voted for the bill as finally adopted, declared that they sought only improvement of the proposed legislation by removal of provisions which they felt had no proper place in the pending measure.

Opponents of the administrator idea still were betting on the Mills bill in the House. In the home stretch, no Senate-House compromise was in the offing.

Services Act—Not content to wait for Congressional fiddling to stop, the armed services have decided to start an integrated research program.

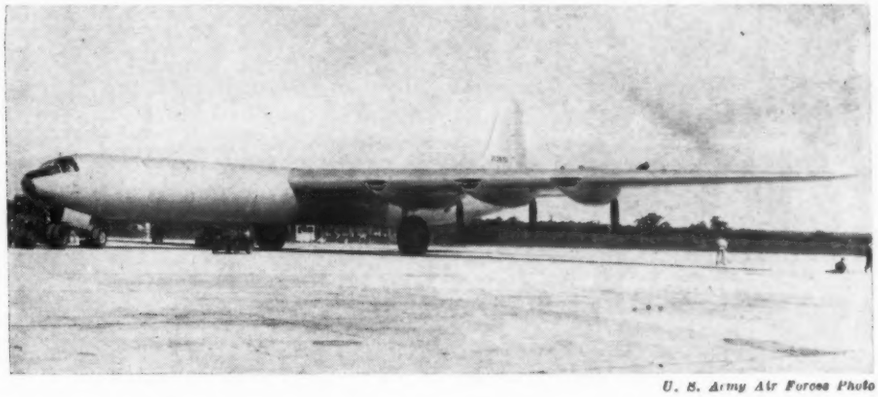
Secretary of War Robert P. Patterson and Secretary of the Navy James Forrestal announced the establishment of a Joint Research and Development Board to carry out "a strong, unified, integrated and complete research and development program in the field of national defense."

Following an organization meeting, Dr. Vannevar Bush was named chairman of the board.

Those who attended the meeting were the Assistant Secretary of the Navy, W. John Kenney; Admiral De Witt C. Ramsey, Vice Chief of Naval Operations; Gen. Carl Spaatz, Commander of the Army Air Forces; Gen. Jacob L. Devers, Commander of the Army Ground Forces, and Dr. Lloyd Berkner of the Carnegie Institution.

Previously established joint boards and committees engaged in research and development will be reconstituted as committees of the new joint board.

The announcement said that the board would not attempt to control joint research and development activities but would leave that function where it now rested within



U. S. Army Air Forces Photo

World's Largest Land-Based Bomber

CONSOLIDATED-VULTEE XB-36 is being readied for its initial ground tests. (Advance artists' sketch of the XB-36 appeared in the May issue of PRODUCT ENGINEERING.) The bomber is a pusher type aircraft powered by six 3,000 horsepower Pratt and Whitney engines. It has a wing spread of 230 feet and a fuselage length of 163 feet. The XB-36 requires a crew of 15 who are accommodated in pressurized compartments.

the two departments. Nor would the board make any material change in actual research, such as pooling of personnel and funds.

The aim of the board will be to provide a freer exchange of information between the services and allocation of research to either the Army or the Navy where there is a joint interest. The main purpose is to abolish or limit overlapping activities.

Besides the Secretaries of War and Navy the board will also keep the Joint Chiefs of Staff and other service agencies informed of research and development projects. In carrying out its mission the board will coordinate budget needs, determine what action is needed to fill gaps in the plans of the two departments, seek the settlement of differences and encourage direct dealing between the Army and Navy, and invite the National Advisory Committee on Aeronautics and other agencies to coordinate research and development activities.

The Latest in Atoms

PRIVATE ENTERPRISE is showing its faith in the peaceful aspects of atomic energy. Notwithstanding the physical bombast of the Bikini atom bomb tests or the verbal fireworks of the United Nations Commission on the Control of Atomic Energy, American business has demonstrated its interest in the future of a great potential industry, by a series of four announcements:

1. Plans have been made for construction within the next few months of a $2,500,000 "pile" at Oak Ridge to be operated for the Army by Monsanto Chemical Company at Clinton Laboratories to conduct experiments in development of electric power through atomic energy.

2. A contract with the General Electric Company to take over operations of the Hanford Engineer Works at Pasco, Washington on next September 1 from the du Pont Company, which will include potentialities of atomic energy as a power source.

3. A 100,000,000-volt betatron has been ordered by the Manhattan Project from the General Electric Company for further experiments in atomic energy. The betatron was ordered for use by the Monsanto Chemical Company at Clinton Laboratories and will be housed at Oak Ridge.

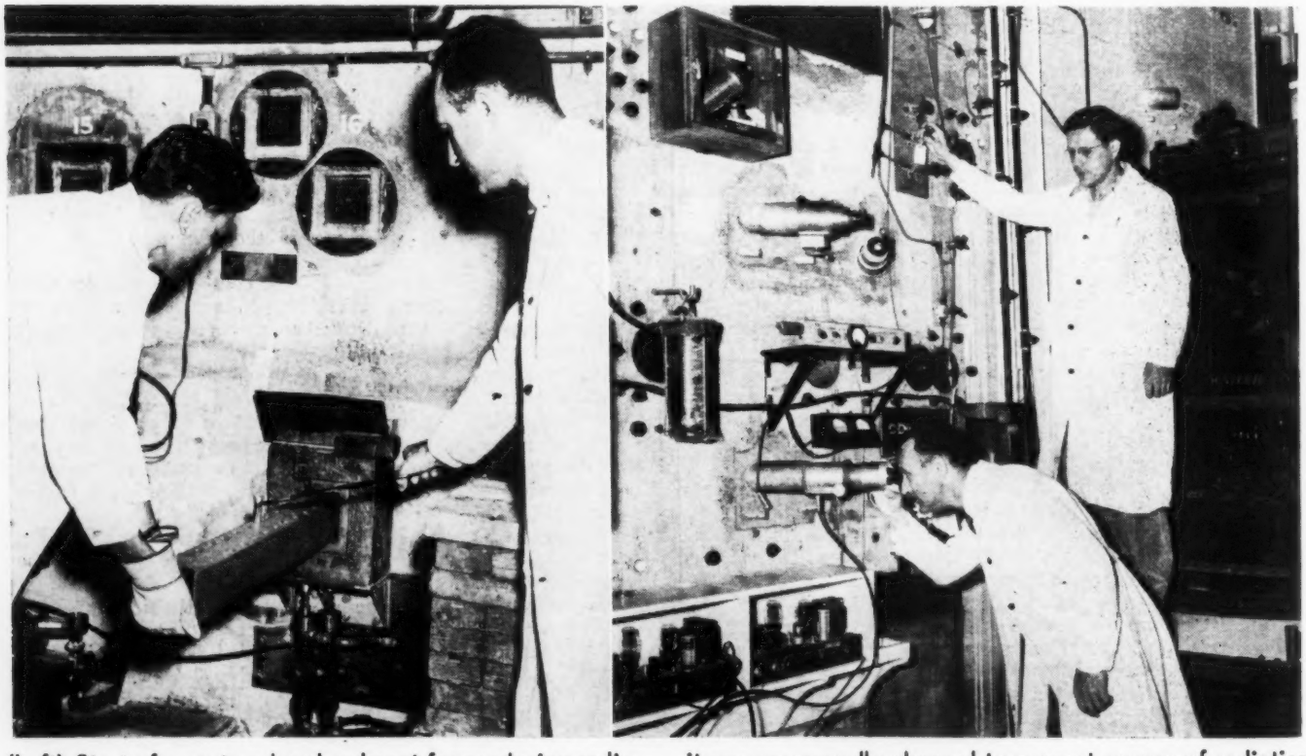
4. Details of a program for the nation-wide distribution of beneficial radioactive isotopes to be produced from the uranium chain-reacting "atom pile" of the Clinton Laboratories, which conducted basic research in development of plutonium atom bomb material during the war, has been announced by the Manhattan Engineer District of the War Department.

Dr. Charles Allen Thomas, vice-president and technical director of Monsanto, will be in charge of the "power plant" project, which is based on the applied research for which Dr. Farrington Daniels, head of the Chicago Metallurgical Laboratory, is largely responsible.

It is not expected that the experimental pile will be an economical power producer or will even provide any clear evidence as to the economics of atomic power. The main thing is to try out a lot of ideas that have been developed. One objective is to forestall foreign development work—primarily British or Canadian—which might lead to applications for U. S. patents on power production devices.

The new power production unit, known as the Daniels pile is similar in principle to the chain reaction piles used at Hanford, Wash., for production of plutonium. These piles produced so much byproduct power, that just keeping them cool raised the temperature of the Columbia River several degrees.

The active agent in the Hanford piles is pure uranium, consisting mostly of the inert isotope, 235. The other essential element is a "moderator" which slows the



(Left). Start of a neutron bombardment for producing radio-isotopes in Clinton chain-reacting uranium "pile". Samples of materials are inserted into the pile to make them radio-active. After the little container filled with the material to be bombarded is fitted into the holes of the graphite carrier block, the block will be pushed into the center of the pile. The pile is not operating when this is done. Concrete protective wall surrounds the pile and the holes through it are are normally plugged to prevent escape of radiation. (Hole No. 16 above in picture). Note use of lead bricks around carrier hole for shielding from residual radiation. (Right). Further steps in the extraction of separate fission elements for experimental uses. Here, Dr. Edward Tompkins demonstrates one of the remote control operations for Dr. Waldo Cohn, who is observing through a periscope the reaction taking place inside the heavy concrete cubicle.

high-speed neutrons emitted during the fission of U-235.

Other essential features of the Hanford piles are: shielding (lead, concrete, or water) to protect operators from the intense radiation produced; control equipment (metal bars which are inserted into the pile to absorb neutrons when it is desired to slow or stop the chain reaction); cooling equipment to dissipate the energy produced.

Under the War Department contract, G. E. will take over from du Pont the operation of the government's $347,000,000 Hanford plant. G. E. had done important research on atomic energy and related fields before the war.

The Hanford Works will be operated by a manager with an advisory committee consisting of H. A. Winne, vice president in charge of engineering policy; Dr. C. G. Suits, vice president and director of the G. E. Research Laboratory, and Dr. Zay Jeffries, vice president and general manager of the G. E. Chemical Department.

The contract with G. E. is similar to the du Pont contract in that the fee for all work to be performed is one dollar. The government will retain control of the plant.

The radioactive isotopes to be made available will be used in research work in fundamental and applied sciences, particularly in biology and medicine. They will be used in two important ways: First as tracer atoms for following the course of atoms in chemical, biological and technical processes; and possibly second, after considerable research, as therapeutic agents for treatments of certain special diseases.

The radioisotopes will be prepared largely in the Clinton Laboratories operated by the Monsanto Chemical Company. The bombardment facilities of the Hanford Engineer Works at Pasco, Washington will also be used insofar as the flexibility of that operation allows. In addition, the Argonne National Laboratory, to be operated for the Army by the University of Chicago under the general guidance of representatives of some twenty-five participating universities, will aid materially in pertinent research.

Distribution will be coordinated and supervised by an Advisory Committee on Isotope Distribution Policy, members of which were appointed by Major General L. R. Groves on the recommendation and nomination by the National Academy of Sciences. To judge radioisotope requests and recommend distribution, the Advisory Committee will have sub-Committees on Allocations and on Human Applications. A technical committee on isotopes composed of representatives from the major laboratories of the Manhattan Project will aid the Project in coordinating production and development of requested isotopes.

The Manhattan District has set up a special Isotopes Branch in its Research Division to administer and coordinate the distribution program in cooperation with the scientific committees nominated by the National Academy of Sciences. Dr. Paul C. Aebersold was named chief of the branch.

Under the program, approximately one hundred radioactive isotopes will be obtainable in varying quantities. Some of the most important of these include Carbon 14, Sulphur 35, Phosphorus 32 and Iodine 131. More than 400 man-made radioactive isotopes are now known. There is at least one such radioisotope for each element. The "Transplutonium" elements, 95 and 96, were recently found in Manhattan Project researches.

A reasonable charge will be made for all isotopes to cover the "out-of-pocket" costs to the government resulting from the additional production incurred by the non-project distribution. Although many isotopes are expensive to produce, especially if desired in a pure form, research-program quantities of important isotopes should not be prohibitively expensive to the average scientific institution.

Infra-Red Electron Telescope Aided Navy's "Invisible" Operations

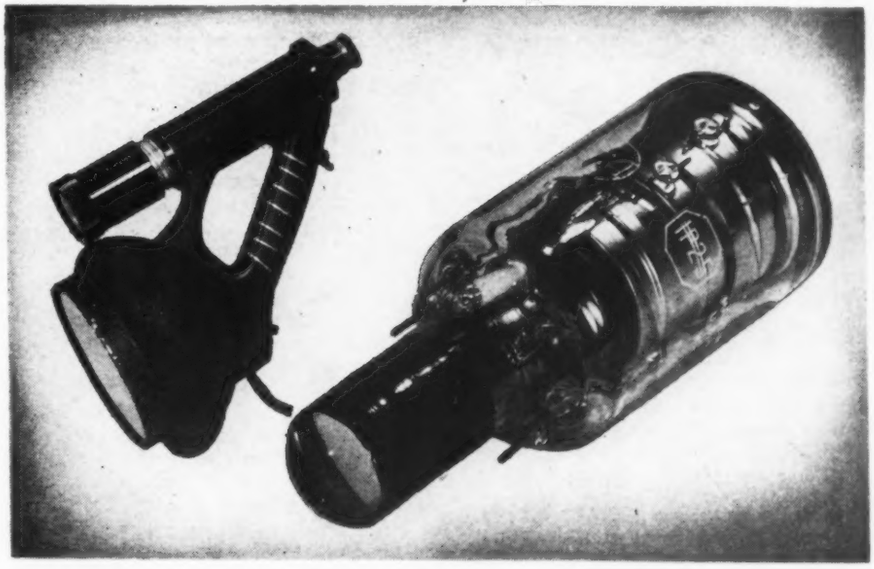
BEACHES IN TOTAL DARKNESS to enemy observers were made clearly visible to U. S. Navy forces through the use of infra-red-sensitive electron telescopes and infra-red floodlights and markers. These instruments enabled the Navy to conduct landing, reconnaissance, and offensive operations as well as code communication by infra-red signal lights, under cover of absolute visual darkness.

The heart of the infra-red electron telescope is a small image tube, developed by the R.C.A. Laboratories, Princeton, N. J., utilizing many of the principles employed in electronic television systems, having a photosensitive surface, but especially sensitive to infra-red radiation. Begun as an outgrowth of research on television camera tubes utilizing visible light, and further developed under a contract with the National Defense Research Committee, the infra-red image tubes were ultimately designed as small as 1 5/8 in. in diameter by 4 1/2 in. long.

The same image tubes used in electron telescopes for the Navy were also the basic element of the Army's recently announced "sniperscope" and "snooperscope," also developed at the RCA Laboratories under the same NDRC contract. The sniperscope was attached to a rifle and served as a gunsight as well as a means of seeing an enemy in darkness, while the snooperscope enabled Army personnel to carry out short range reconnaissance operations under security of complete visual darkness.

Basically, the telescope consists of an objective lens for forming upon the sensitive cathode of the tube an infra-red image of the scene being viewed, the tube itself, and an eye-piece. The tube consists of a semi-transparent photosensitive cathode which is ultra-sensitive to infra-red radiation, an electrostatic electron lens system, and a fluorescent screen. The light-gathering objective lens system concentrates the received infra-red image on the photocathode, releasing from the latter a pattern of electrons which conforms to the original image. An electron optical system, consisting of an electrostatic field focuses the electrons coming from the photocathode and accelerates them toward the fluorescent viewing screen where a visible reproduction of the original image is formed. The reproduced image is then viewed through a telescopic eye-piece, in the same way an ordinary telescope is used.

To actuate the electron image tube, a high



"Snooperscope". Infra-red image tube is in monocular cylinder on top of handle. Spot light which transmits only infra-red light is at lower end of grip. (Right) RCA Image tube is the eye of the "Sniperscope", "Snooperscope" and other devices for observing scenes in total darkness.

voltage electrical power supply is necessary. The required high voltage was obtained from a small lightweight unit, powered by two ordinary flashlight cells, developed by RCA engineers.

In peacetime, infra-red finds application in burglar alarms, microscopy, spectroscopy and examination of undeveloped film in total darkness.

MEETINGS

August 19-23.
National Association of Power Engineers—Annual Convention and Power Show, Hotel Cleveland, Cleveland, Ohio.

August 20-23
American Mathematical Society—Summer meeting and Colloquium, Cornell University, Ithaca, N. Y.

August 22-24
Society of Automotive Engineers—West Coast transportation and maintenance meeting, New Washington Hotel, Seattle, Washington.

September 9-14
American Chemical Society—National meeting and Chemical Exposition, Chicago, Ill.

September 16-20
Instrument Society of America—First national instrument conference and exhibit, William Penn Hotel, Pittsburgh, Pa.

September 30-October 3
American Society of Mechanical Engineers—Fall meeting, Hotel Statler, Boston, Mass.

RECENTLY ELECTED SOCIETY OFFICERS

American Institute of Electrical Engineers, 33 W. 39th St., New York 18, N. Y. President—J. Elmer Housley, Aluminum Company of America; Vice President—E. W. Davis, Cambridge, Mass.; Treasurer—W. I. Slichter, Schenectady, N. Y.

American Society for Testing Materials, 1916 Race Street, Philadelphia 3, Pa. President—Arthur W. Carpenter, B. F. Goodrich Co.; Vice President—R. L. Templin, Aluminum Company of America.

American Gear Manufacturers Association, Empire Building, Pittsburgh 22, Pa. President—Thomas J. Bannan, Western Gear Works; Vice President—Raymond B. Tripp, Ohio Forge and Machine Corporation.

American Rocket Society, 29 W. 39th St., New York 18, N. Y. President—Lowell Lawrence, Reaction Motors, Inc.; Vice President—Roy Healy; Secretary—G. Edward Pendray; Treasurer—Samuel Lichtenstein.

U. S. Navy Photograph

Radical "Flying Pancake" Design Announced

A CONVENTIONALLY-POWERED plane that can hover like a helicopter, the XF5U-1 is revealed as one of the most radical designs in aviation history. Built for the Navy by the Chance Vought Aircraft division of the United Aircraft Corp., the new plane consists essentially of a round wing with tractor propellers at the outer edges.

Salient features of the XF5U-1 is its amazing speed range. While designers for years have not been able to do better than a 1 to 4 ratio of landing speed to top speed, the Vought aircraft ranges from 40 to 425 mph. with standard engines; 20 to 460 mph. with water injection engines, and 0 to 550 mph. with gas turbine power plants.

The last ratio, focuses the spotlight on the airplane's other distinctive characteristic: ability to hover on its propellers. This is made possible by a specially-articulated type of blade, similar to that used on helicopters. These propellers were developed for the XF5U-1 by Hamilton Standard, another United Aircraft subsidiary.

By standing the plane on its tail, the pilot will be able to suspend in the air, his forward speed depending on the power available, 0 mph. requiring the greatest power. In this position, as the propeller blade moves forward, it maintains a desired angle; as it sweeps aft it flattens out as in a helicopter rotor.

The XF5U-1 is expected to be given its test flight in September. The practicability of the design, however, has been proven in flights of an earlier version, the V-173, which first flew in 1942. This, although a full-scale model, was constructed of wood and fabric and used low-power engines.

Other features of the XF5U-1 include: Power by two Pratt & Whitney R-2000 Twin Wasp engines developing 1,350 h.p. at 2,700 r.p.m. for takeoff. These engines are "buried" in the wing on either side of the cockpit and drive the propellers at the tips through right-angle transmission systems which include approximately 5:1 propeller reduction gearing. Special clutches permit either engine to drive both propellers.

Controls located on the wing trailing edge including twin vertical fins and rudders and special elevon-type horizontal surfaces extending outboard from the rear of the wing. These latter consist of "elevators" for longitudinal control and "ailerons" provided by differential movement of the stabilizer portions, which are controllable.

Jet Engine Delays Seen by Developer

THE MAN WHO MAY BE SAID to have invented the jet turbine engine, Air Commodore F. J. Whittle, of the Royal Air Force, gave to a meeting of the New York Section of the Society of Automotive Engineers on July 15 at the Hotel Pennsylvania a picture of his own part in developing these revolutionary power plants and a prediction of their immediate future.

It will be five to ten years, he said, before details of their construction have been solved to the point where they will replace the conventional reciprocating engine in all but light personal planes. It will be that long, he said, before people need generally to become accustomed to the thought of flying around the world at more than 500 miles an hour and at altitudes above 40,000 feet.

Commodore Whittle does not subscribe wholly to the belief that the Germans were far ahead of us in this field, he said, asserting that they started the development at the same time he did in England, in the early Thirties. They did, however, have a jet engine in the air first and had more jet fighters than anyone else before the war ended, he admitted. But he pointed out that the development of their first jet had been discontinued and that the Whittle engine was the demonstrated "father" of a long series of useful successors.

Future developments, the Commodore said, will proceed in many directions as compressors, combustion methods and materials to withstand the high temperatures involved

are improved to a high degree.

Among the directions in which he said he expected immediate development were in the application of jet turbines to heavy aircraft for speeds in the 350-mile-an-hour category. From 350 miles to 500 miles, he said, he thought the jet turbine would best be applied. But for speeds above 600 miles an hour and for flight at and above 40,000 feet, categories in which both British and American jet fighters already are operating, he foresaw the almost immediate and complete displacement of the reciprocating engine with propeller by the pure jet-turbine.

After a series of lectures in the United States, Commodore Whittle will return to England to head research at a series of important new RAF stations. He is technical adviser to the Ministry of Supply for Air.

A.S.T.M. Annual Meeting Features Materials and Assemblies

THE 1946 A.S.T.M. ANNUAL MEETING held in Buffalo indicated the intensive interest in the Society's rapidly expanding work on specifications and tests for materials and methods of evaluating assemblies of these materials in various forms. There were eight symposiums with technical papers covering a wide diversity of fields. The topics included: Symposiums on Bearings, Gas Turbine Materials, Fatigue, Spectroscopic Light Sources, Oil Procurement Practices, Testing Parts and Assemblies, pH Measurements, Atmospheric Weathering of Corrosion-Resistant Steels, and Freezing-and-Thawing Tests of Concrete. In addition to these symposiums, there were other sessions devoted to effect of temperature on metals, cement and concrete, plastics, nonferrous metals, bituminous materials, and others, 25 separate sessions being required to provide adequate time for all the papers and reports.

The Symposium on Testing Parts and Assemblies was a joint one with the Society for Experimental Stress Analysis which held its meeting in Buffalo during the week of the A.S.T.M. meeting.

One of the meeting features was the Edgar Marburg Lecture by Dr. J. J. Mattiello, Technical Director, Hilo Varnish Corp., on the topic "Protective Organic Coatings as Engineering Materials." The Nineteenth Award of the Charles B. Dudley Medal was made to H. R. Copson, The International Nickel Co., Inc.

"Plastic Plug" Renews Oil Well Production

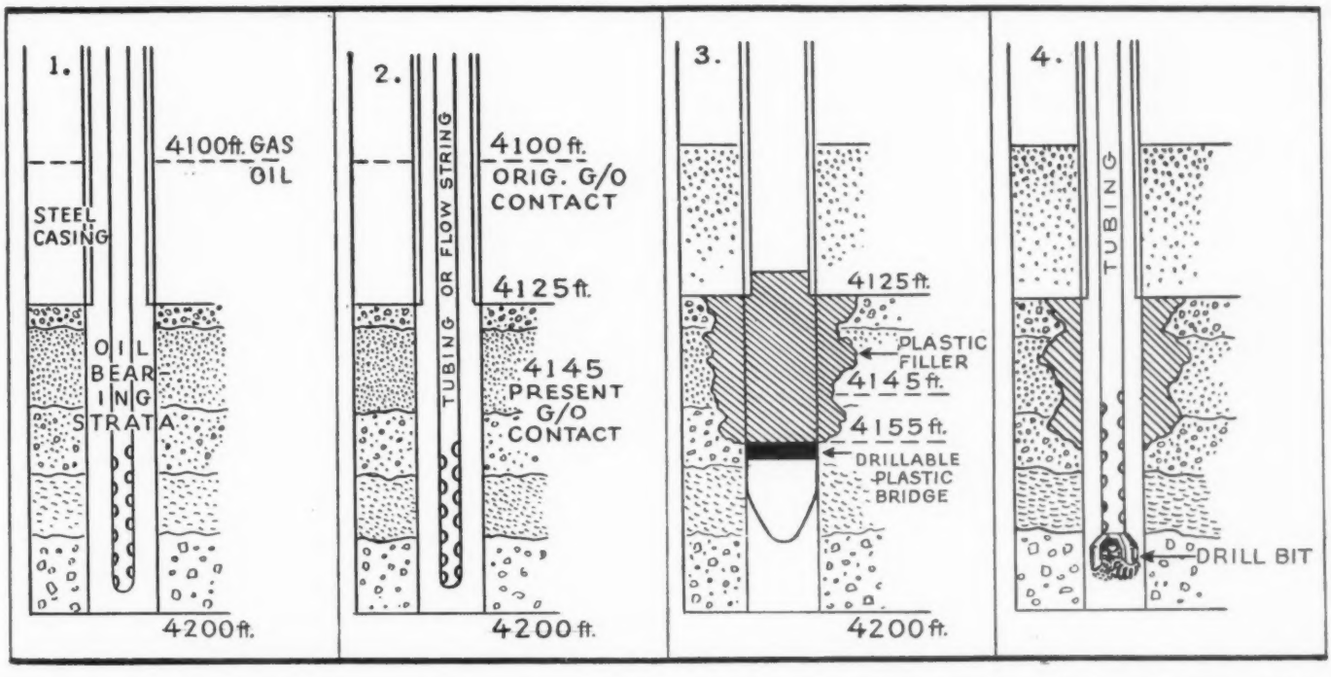
A PLASTIC that puts "new life in old oil wells" was announced recently by Monsanto Chemical Company of St. Louis and Oilwell Chemical Service Company of Fort Worth, Texas. Tests in the west Texas oil field, according to the two companies, indicate the "plastic plug" development will return the average well to its normal production after its allowable output has dropped to as low as one barrel daily because of the intrusion of unwanted natural gas.

The plastic, phenolic resin, has been used as deep as 11,500 feet to seal wells against the intrusion of natural gas. Unlike previously-used cement, containing solids held in suspension, the resin is introduced into the shaft as a true liquid, which permits maximum penetration and hardens after it permeates the porous rock through which gas has been filtering.

The application increases the allowable petroleum, rather than the available petroleum. This is because, under state laws, producers must maintain a close relationship between the amount of oil and natural gas that can be withdrawn in one day. For example, the allowable might be 50 barrels of oil daily, with a maximum of 2,000 cubic feet of natural gas per barrel, or a total of 100,000 cubic feet of gas opposed to 50 barrels of oil.

A seven-man crew can normally complete the remedial operation in 36 hours.

The resin is in volume production at Monsanto's Plastics Division, Springfield, Massachusetts.



1—New oil well, 4200 feet deep, with gas-oil contact point at 4100 feet. 2—After six years' pumping, gas-oil contact drops to 4145-foot level and gas intrudes via porous rock through which oil previously filtered. 3—Flow string (perforated metal inner tubing) is removed and drillable plastic bridge (with remotely-controlled "dogs" which bite into walls) is spotted at 4155-footlevel. A syrupy phenolic resin is introduced into shaft and pressured into porous rock. It hardens in about eight hours. 4—Drill attached to re-inserted flow string reams out hole through plastic and disintegrates bridge. Meanwhile, the hardened phenolic resin seals off the unwanted natural gas from the oil pay. This allows normal production to be resumed. The drill bit remains in the well.

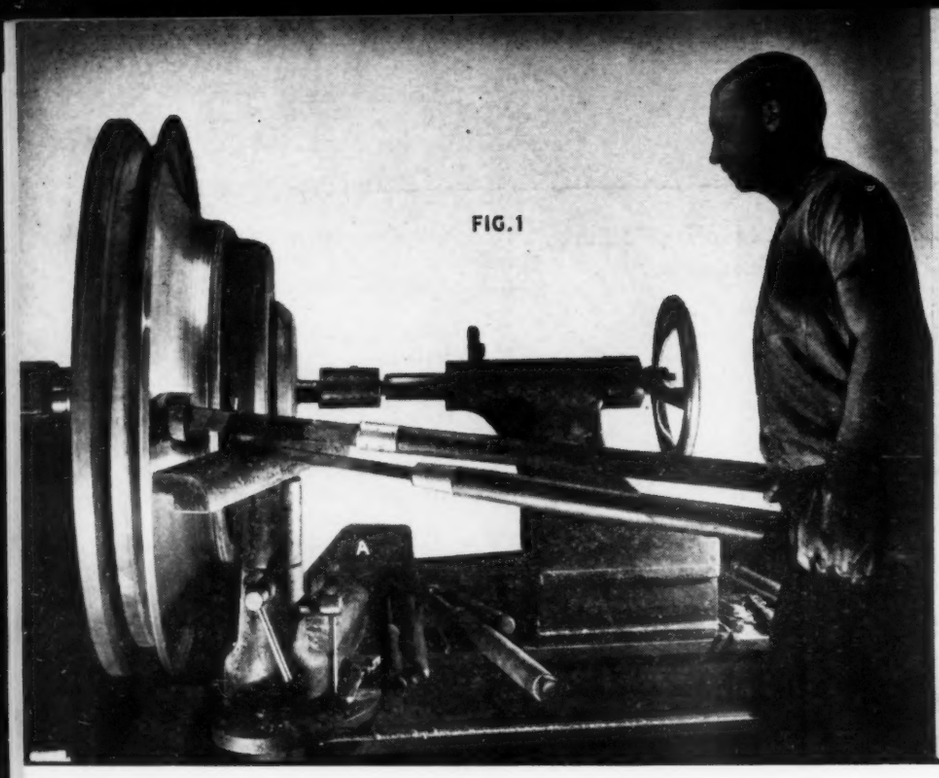


FIG. 1

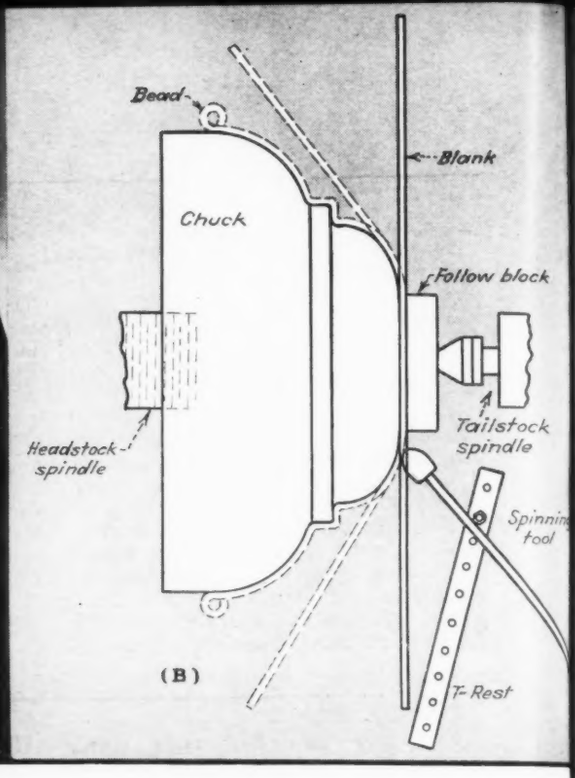


FIG.

Design of Parts to Be Formed By Metal Spinning

C. J. HOLINGER
Perry Metal Products Company

General types of parts suited to the spinning process, properties governing the selection of materials and design tolerances are discussed, together with information pertaining to chuck design. The metal spinning process, its economy and rates of production are compared with those of deep drawing on power presses.

SINCE there are few metal spinning shops, engineers and designers have had little chance to learn how to design parts for production by the spinning process. Metal spinning is a deep drawing process in which a circular piece of sheet metal is cold worked to the shape desired by forming it against a chuck revolving in a lathe.

The chuck determines the shape of the part; hence those dimensions of the part that conform to the surface of the chuck should receive special attention in the design of the part. They should be given on the drawing so as to define both the part and the chuck. The thickness and ductility of the metal are also of major importance in the design of parts to be spun. In considering these factors the designer may well take a cue from a study of the spinning process.

Deep drawing in the spinning process is totally different from deep drawing in a press. There is no die—only a punch. There is no pressure pad, the work being its own pressure pad. The operator is at least as important as the toolmaker.

Metal spinning is an art or trade that retains the elements of craftsmanship. The work is physically hard. The spinner has no carriage stops or graduated dials on his lathe; he is obliged to rely on his sense of touch and on acquired skill in coordinating hand and eye during the spinning operation. His job is to make the part according to the drawing; the designer's job is to make the spinner's work as easy as possible by a design suited for spinning the part.

Spinning Process

In spinning, the chuck of the desired shape is attached to the headstock of the lathe. The circular blank of sheet metal is placed against the end of the chuck and supported by a follow block fixed to the tailstock, Fig. 1(A) and (B). The follow block revolves with the blank.

Pressure is applied to the blank by the spinning tool, which is usually steel or brass. The necessary leverage is obtained by the tool resting against a fulcrum inserted in a T-rest near the rotating blank. The T-rest has a series of holes so that the position of the fulcrum can be changed as required. The blank is forced first against the chuck in the area nearest the axis of rotation, progressing successfully to areas further from the axis until the blank fits the entire face of the chuck. Finally a smoothing tool may be applied to straighten out ridges and give a smooth finish.

Although the process appears simple, considerable skill is necessary to obtain a shell of uniform thickness or to confine the shell

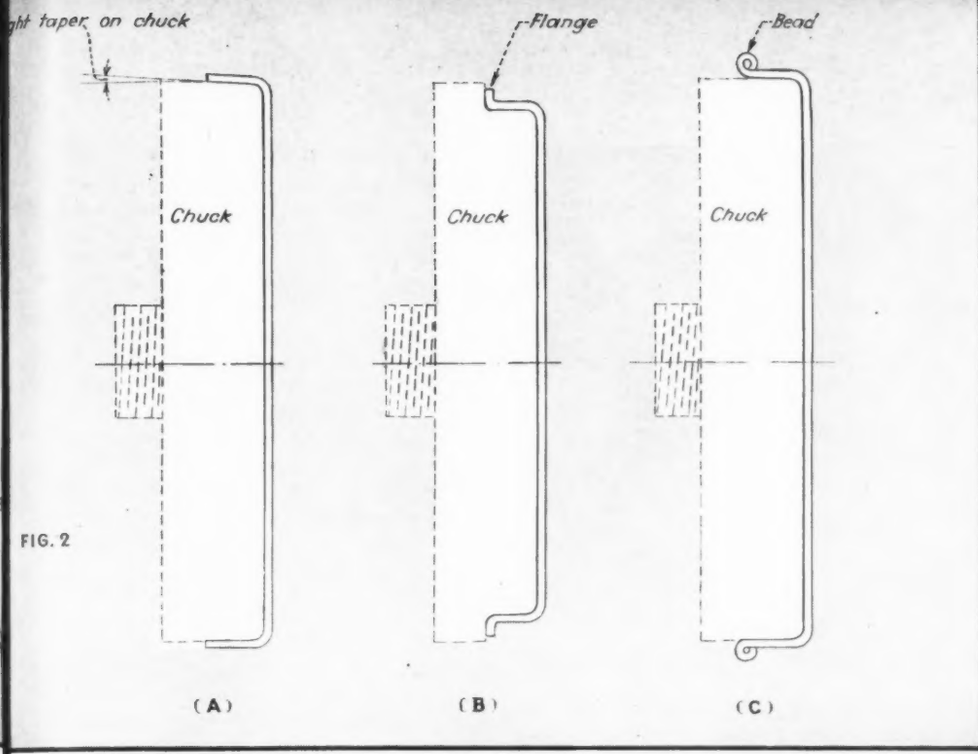


FIG. 2

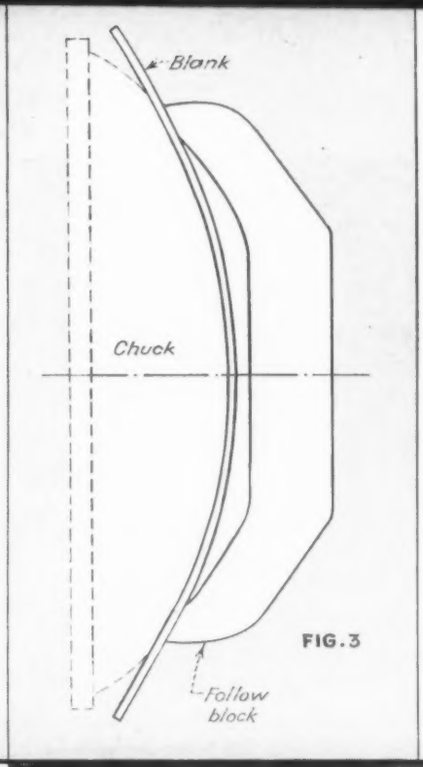


FIG. 3

Fig. 1—(A) Metal spinning is an art requiring an operator with good mechanical sense, strength of arm, and acquired coordination of hand and eye. (B) To spin metal parts, the blank, usually clamped between a chuck and a follow block, is cold-worked by hand operated spinning tools to fit the revolving chuck.

Fig. 2—At least a slight taper is necessary to prevent locking the part onto the chuck. Radii at corners and fillets should be as large as practicable.

Fig. 3—Follow blocks on large shells often are required to hold the work near the area to be spun. Generally more than one follow block also is necessary.

to a prescribed length. Forcing the metal in only one direction produces an elongated shell with thin walls. Therefore, back strokes of the spinning tool as well forward strokes are necessary. The forward strokes (toward the headstock) thin the the metal as the shape develops and the back strokes thicken it again.

Design Fundamentals

In general a shell can be spun on any portion of a solid of revolution so long as the radii of the portion continuously increase with distance along the axis of rotation. The radii must increase, else the shell cannot be removed from the chuck. Wood chucks are practical for shells with cylindrical sides, like Fig. 2, if the design permits a taper on the sides of from 3/8 to 1/2 in. per ft. Where tapering sides are not permissible, a steel chuck is required.

The center of a shell can be marked for drilling or layout purposes by removing the follower block and jamming the tail stock against the rotating shell. Large holes can be cut about the shell center by a swing cutter operated in the manner shown by Fig. 4. Accuracy on the diameter of such holes can be held to a tolerance of from 0.003 to 0.005 in. if necessary.

Sharp edges can be beveled, flanged as in Fig. 2(*B*) or beaded as in Fig. 2(C). The bead will seldom have a true circular section and will vary from shell to shell because a bead usually is spun "in the air," that is, without the aid of a chuck to spin it against. Nothing can be done on the lathe to improve the flatness of bottom of a flat-bottomed shell of the type illustrated in Fig. 2. If the sheet material is not flat enough, the drawing should indicate that the blank is to be flattened before spinning.

Most shells have a small flat on the bottom but for those pieces that have no flat bottom it is necessary to start with a small follow block, then change to a larger follow block. The larger block holds the shell at a greater diameter and keeps the metal from backing off the chuck.

Several follow blocks may be required for large pieces, depending upon the thickness and type of material. Shells of 2S aluminum that are less than 15 in. in diameter can be spun with one follow block. Another follow block can be added for each additional 15 in. in diameter. Large follow blocks are designed to bear against the shell near the periphery only, Fig. 3. If the part is to have no flat at all, the small flat left under the first follow block is spun down after the rest of the shell is finished.

Reflectors, lamp bases, lighting fixtures, air deflectors and many similar articles can

Fig. 4—Large holes can be cut from the center of soft metal shells by swing cutters used in the manner shown here.

be designed for production by spinning. Many covers for machines and equipment are produced by spinning for use in places where curved surfaces are wanted for appearance. Rigidity of the part can also be incorporated into covers designed for production by spinning. Fig. 6 is an illustration.

The shape of central portion of this part might have been conical, elliptical, parabolic, circular, but such shapes would have required the use of more than one follow block. Instead, the shallow cup at the center,

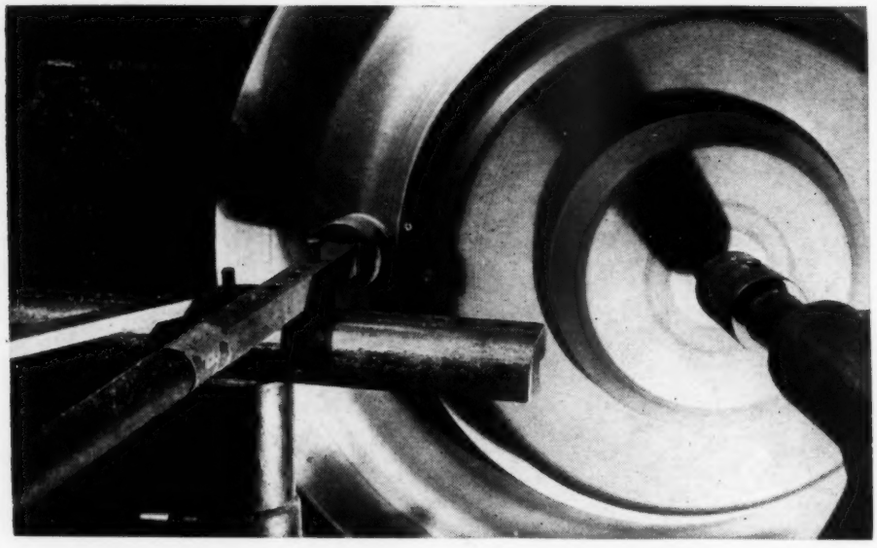


Fig. 5—Close-up view showing double anchors whereby the beaded tool can be applied to the work on both forward and backward strokes.

which is spun first, admits the use of one follow block and keeps the metal from backing off while the rest of the part is spun.

A depth of cup of about four times the metal thickness is sufficient. Proportions such that diameter *d* is 15 to 25 percent of diameter *D* are practical. The channel shape, exterior to the conical portion *D*, gives rigidity to the part. Although the 5 deg. cone angle is not a minimum, it is on the safe side and will keep the cone from reversing like an "oil can."

So far the applications have all been shallow shells such as might be made in one draw on a deep drawing press. Just as deep shells have to be drawn in stages so shells need to be spun in stages, often with an annealing operation between stages. The preliminary operations in spinning are called "breaking down" operations. Sometimes they are as important as the final spinning operation. Although the finished piece retains no evidence of the breaking down operations, usually it is impossible or uneconomical to spin without them.

The shell spun onto the Masonite chuck of Fig. 7 is an illustration. The chuck is 24 in. in diameter and 26 in. deep. Breaking down is necessary to give the rim of the blank some stiffness while the spherical part is being spun. The workability of an aluminum blank can be restored by heating with a blow torch the partly formed blank while it is revolving. Where annealing is necessary between break down operations, better practice is to remove the part from the lathe and anneal it in an oven or salt bath. Small parts of aluminum can be immersed in a molten nitrate bath at 650 to 800 deg. F. The heating time varies from one to several minutes, depending upon the thickness of the sheet.

There are two ways of making a part that has a reversal of shape, such as the part shown in Fig. 8(A). If the center cylinder is deep and small, it is spun first as in Fig. 8(*B*). This displaces enough material to make the inner cylinder. Then the partly formed piece is reversed and the outer cylinder spun against a chuck on the opposite side of the part.

If the part has a shallow inner cylinder and no bottom, the center can be sunk so that the job can be spun on a single chuck. Fig. 8(C). The steps are: First, spin down the outside flange. A large follow block must be used during this operation. Second, cut out the center. And third, sink the inside flange. Some device other than a follow block is required to hold the shell while the inside flange is spun. One way is to use an almost perpendicular outside flange. Another is to lock the edge of the outer flange against the chuck and cut off the locking portion after the inside flange is finished.

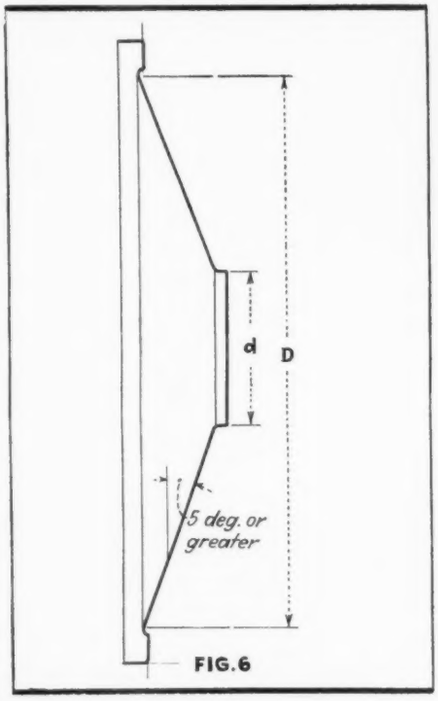


Fig. 6—Parts can be designed for rigidity as well as for ease of manufacture by spinning.

Ductility is an essential property of any material that is used in making parts by spinning. From the spinner's standpoint 2S aluminum, which practically does not work harden, is one of the best materials. Soft copper also has low work hardening. It is preferred by some to aluminum. Of the brasses, 70 Cu-20 Zn has high ductility. It must be spun carefully with fast, accurate, heavy strokes to get the maximum draw between annealings.

Stainless steels are more difficult to spin than aluminum, brass and copper. Although stainless steels have high ductility, they cannot be spun without annealing to anywhere near the depth to which aluminum and copper is spun. Frequent annealing should be specified to avoid cold working the material during spinning to the extent that cracks are formed.

Low carbon steel can be spun to wider limits than the stainless steels. Annealing should be conducted with as little oxidation of the surface as possible. An annealing temperature of about 2,000 deg. F. is customary.

Fig. 7—Masonite chucks deform less under heavy spinning and also change less with aging than wood chucks.

Bronze, magnesium, Monel, zinc, nickel, nickel silver and other alloys can be used for spun parts. Monel and nickel do not flow as much as the softer metals so that the designer can make the spinner's job easier by specifying a blank slightly larger than that required for the softer metals. Annealing temperatures for Monel and nickel vary from 1,550 to 1,800 deg. F. Nickel need be kept in the annealing furnace only about half as long as Monel, assuming equal temperatures.

Metal parts can be spun successfully in all sizes from 1/2 to over 100 in. in diameter.

Usual commercial tolerances are plus or minus 1/16 in. on large parts, 1/32 in. on parts of medium size, and 1/64 in. on small parts. Tolerances of plus or minus 0.005 in. can be maintained where necessary.

In general, sharp bends and fillets of small radius are to be avoided. Corners on wood chucks less than 1/4 in. radius fail under pressure of the spinning tool. Sharp corners on plastic and Masonite chucks chip and break out. Where small radii are necessary steel chucks are recommended. At best, the metal is always thinned more at corners by spinning so that radii should always be as large as the design of the part will allow.

Taper on wood chucks should be at least 3/8 in. per ft. to insure removing the work from the chuck. About 3/32 in. per ft. is the minimum taper for Masonite chucks, and 1/32 in. per ft. is a practical minimum for steel chucks. A part like Fig. 6 needs no taper at all on the cylinder of diameter d because the working of the cone will pull the shell off the step just enough to keep the shell from locking.

Aluminum and soft copper can be spun economically in thicknesses as great as 1/4 in. A thickness of 3/16 in. is about the limit for low carbon steel and 1/8 in. for the stainless steels. Aluminum sheet only a few thousandths thick has been spun on small work up to 4 in. blank diameter, but blanks 0.010 in. thick are easier to spin. The majority of parts for production by spinning are made of blanks from 0.025 to 0.050 in. in thickness.

Maple is commonly used for wood chucks. Beech wood is also used, particularly for small lot production. Lignum vitae is an excellent wood for chucks, but it is expensive and often not available.

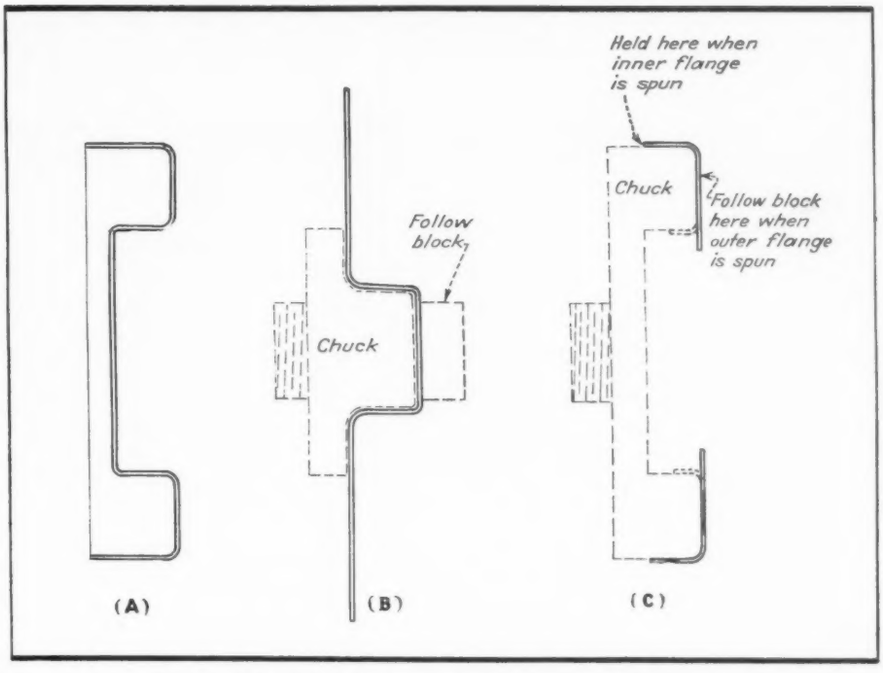


Fig. 8—(B) and (C) illustrate two ways of spinning a part like (A), depending upon the dimensions of the part.

Metal chucks may be made of steel, aluminum alloys or Zamak. Collapsible metal chucks have been designed, which permit the spinning of shapes that would otherwise be impractical because the parts could not be removed from them. The cost of such chucks, however, is generally prohibitive.

A primary reason why the spinning process is so important as a method in metal fabrication is its low cost on small-quantity production. Power press methods are much faster, give more uniform product, and are more economical for the production of parts in large quantities. If thousands of the same part are to be made, then the designer would choose the press method of cold-working the part to dimensions. If only 50 or a 100 were required, he would specify production by metal spinning. Where production by spinning stops and press production begins is debatable. Perhaps 5,000 parts is an average dividing point.

Motor Driven Drawing Board

LARGE BOARDS that permit full scale lay-out have been devised by the Landis Tool Company. A board 6 ft. x 16 ft. can be raised and lowered by a reversing gear motor operating by limit switches which are actuated by the directional pressure of the engineer's hand on a control rod at the bottom of the board.

Pulling up on the rod will raise the board and pushing down will lower it. The operating motor is small, as the board's weight is counter-balanced and the motor actually creates an unbalance in either direction thus making for easy movement of the board.

Isolation of Vibration In Spring Mounted Apparatus—II

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Procedure for calculating linear and angular displacements, natural frequencies, and rocking and torsional modes of resiliently mounted apparatus. Exact and approximate equations are derived for the spring mounts in any position either above or below the center of gravity and not necessarily symmetrical with any major axis.

THE RADIUS of gyration is the main factor not immediately available when it is desired to estimate the natural frequencies of mounted apparatus by the methods discussed in Part I of this article, see page 106, July, PRODUCT ENGINEERING. Fortunately, a rather rough approximation of the radius of gyration will often be sufficient for estimating natural frequencies.

A rough guess is to assume a reasonable distribution of the mass of the equipment and then apply the usual formulas as derived in elementary mechanics.

If the equipment is available, the radius of gyration r can be determined experimentally by the method outlined by W. Ker Wilson, "Practical Solution of Torsional Vibration Problems." The equipment is hung by two wires equidistant from the center of gravity as shown in Fig. 8, and the period of oscillation as a torsional pendulum rotating in a horizontal plane is determined by counting and timing with a stop watch.

Where

R = radius from the center of gravity to the suspension wire, in.
L = length of suspension wire, in.
T = period of torsional oscillation, sec.

the radius of gyration r about the axis parallel to the suspension wires and passing through the center of gravity is

$$r^2 = \frac{386 R^2 T^2}{4\pi^2 L} \qquad (26)$$

This experimental method of determining the radius of gyration is precise, but is often an unnecessary refinement. Since the primary purpose in mounting is to avoid resonant frequencies by a rather large margin, a rough estimate of the natural frequencies is often sufficient. Hence in many designs, computation of the radius of gyration by approximate formulas is often justified.

CASE II — NON-SYMMETRICAL BOTTOM MOUNTS. The equations derived in Case I (July, PRODUCT ENGINEERING) are predicated on the assumption that the mounts were symmetrically placed about the vertical axis through the center of gravity. For bottom mounts non-symmetrically placed, even if they are not theoretically correct, the equations will usually give reasonably accurate results by using in place of a^2, b^2 and d^2 the average of the squares of the distance to the mounts in the plane being considered. For example, if the box of Fig. 9 is considered for only the bottom mounts, the charts (see Reference Book sheets in July, PRODUCT ENGINEERING) can be used to approximate the natural frequencies. In the yz plane, the average value of b^2 is

$$b^2 = \frac{3.38^2 + 8.62^2}{2} = \frac{85.73}{2} = 42.86$$

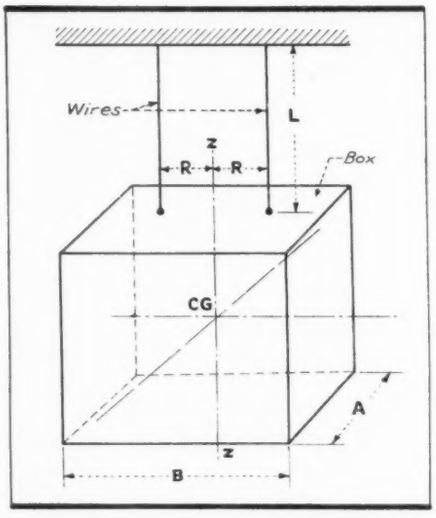

Fig. 8—Arrangement for experimentally determining the radius of gyration.

GENERAL CASE: FORCE AND MOMENT EQUATIONS. For the general case, the mounts may have any location either above or below the center of gravity, and not necessarily symmetrical with any major axis. A general solution is rather complicated and should be avoided, if possible, by placing the mounts in a symmetrical pattern.

As in the previous cases, the general equations for forces and moments are set up by equating the inertia forces or torques plus the external forces or torques to zero. For the box shown in Fig. 10, the external force F_0 applied along the x axis at the center of gravity will produce the displacements and rotations shown.

Where:

C_x = horizontal stiffness in the x direction of any mount
Z = distance from that mount to the center of gravity parallel to the z axis.
Y = distance from that mount to the center of gravity parallel to the Y axis.
δ_x = deflection along x axis
α = angle of rotation in xz plane
β = angle of rotation in xy plane
m = mass
ω = exciting frequency, radians per sec.

For the xz plane, the forces acting in the x direction are

$-F_0$ = applied force
$\delta_x \Sigma C_x - \alpha \Sigma C_x Z$ = spring force caused by motion in the xz plane
$-\beta \Sigma C_x Y$ = spring force caused by rotation about z axis
$-m\omega^2\delta_x$ = inertia force

These forces give the equation

$$F_0 = (\Sigma C_x - m\omega^2)\,\delta_x - \alpha \Sigma C_x Z - \beta \Sigma C_x Y \qquad (27)$$

This is the first of the six necessary force and moment equations for the general case. In determining the numerical values of terms such as $\Sigma C_x Z$, care should be taken to observe the sign convention adopted for the distances from the mount to the center of gravity.

Where

K = vertical stiffness of one mount
X = distance to that mount from the center of gravity parallel to the x axis
I_y = mass moment of inertia about the y axis
γ = angle of rotation about x axis
δ_z = deflection along z axis

for rotation in the xz plane the following moments will act

$\alpha \Sigma K \delta_z^2$ = moment resulting from vertical deflection of the springs caused by rotation in the xz plane

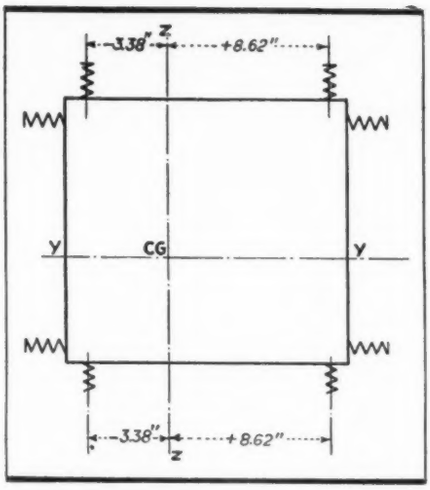

Fig. 9—Test box mounted unsymetrically in the yz plane.

$\alpha \Sigma C_x Z^2$ = moment resulting from horizontal deflection of the springs caused by rotation in the xz plane

$-\delta_z \Sigma C_x Z$ = moment resulting from horizontal deflection of the springs caused by deflection along the x axis

$\beta \Sigma C_x YZ$ = moment resulting from horizontal deflection of the springs caused by rotation about the z axis

$\delta_z \Sigma KX$ = moment resulting from vertical deflection of the springs caused by movement parallel to the z axis

$\gamma \Sigma KYX$ = moment resulting from vertical deflection of the springs caused by rotation in the yz plane

$-I_y \omega^2 \alpha$ = inertia torque

The summation of these moments lead to

$$M_y = 0 = (\Sigma KX^2 + \Sigma C_x Z^2 - I_y \omega^2) - \delta_z \Sigma C_x Z + \beta \Sigma C_x YZ + \delta_z \Sigma KX + \gamma \Sigma KYX \quad (28)$$

The derivation of the four other force and moment equations is similar. They are

$$F_y = 0 = (\Sigma C_y - m\omega^2)\,\delta_y - \gamma \Sigma C_y Z + \beta \Sigma C_y Z \quad (29)$$

$$M_z = 0 = (\Sigma C_y X^2 + \Sigma C_x Y^2 - I_z \omega^2)\,\beta - \delta_x \Sigma C_x Y + \alpha \Sigma C_x YZ + \delta_y \Sigma C_y X - \gamma \Sigma C_y XZ \quad (30)$$

$$F_z = 0 = (\Sigma K - m\omega^2)\,\delta_z + \gamma \Sigma KY + \alpha \Sigma KX \quad (31)$$

$$M_x = 0 = (\Sigma C_y Z^2 + \Sigma KY^2 - I_x \omega^2)\,\gamma - \delta_y \Sigma C_y Z + \delta_z \Sigma KY - \beta \Sigma C_y XZ + \alpha \Sigma KXY \quad (32)$$

Approximate Solution for General Case. The solution for the completely general case consists in solving a sixth order determinant using the coefficients of δ_x, δ_y, δ_z, α, β, and γ, in Equations (27) to (32). Since that is not too practical, it is expedient to search for approximate solutions.

If the last term in Equation (27) and the last three terms in Equation (28) are neglected, two equations in δ_x and α will be obtained. Proceeding as before, the approximate natural frequencies for the xz plane will be those for which the following determinant is zero:

$$\begin{vmatrix} (\Sigma C_x - m\omega^2) & -\Sigma C_x Z \\ -\Sigma C_x Z & (\Sigma KX^2 + \Sigma C_x Z^2 - I_y \omega^2) \end{vmatrix}$$

Expanding this determinant and solving for the rocking frequencies, and letting

$$\varphi = \frac{\Sigma C_x Z^2/K + \Sigma X^2}{r^2_y}$$

and $\theta = \Sigma C_x / K$

the following approximate equation for the rocking frequencies in the xz plane is obtained.

$$\frac{\omega}{\sqrt{K/m}} = \left\{ \frac{1}{2}\left[\theta + \varphi\right] \pm \left[1/4\,(\theta+\varphi)^2 - \theta\varphi + \frac{(\Sigma C_x Z/K)^2}{r^2_y}\right]^{1/2} \right\}^{1/2} \quad (34)$$

Similarly letting

$$\psi = \frac{\Sigma C_y Z^2/K + \Sigma Y^2}{r^2_x}$$

and $\rho = \Sigma C_y / K$

the rocking frequencies in the yz plane are approximated by

$$\frac{\omega}{\sqrt{K/m}} = \left\{ \frac{1}{2}\left[\rho + \psi\right] \pm \left[1/4\,(\rho+\psi)^2 - \rho\psi + \frac{(\Sigma C_y Z/K)^2}{r^2_x}\right]^{1/2} \right\}^{1/2} \quad (35)$$

The approximate torsional frequency is given by

$$\frac{\omega}{\sqrt{K/m}} = \left[\frac{\Sigma C_y X^2/K + \Sigma C_x Y^2/K}{r^2_z}\right]^{1/2} \quad (36)$$

The approximate vertical frequency is

$$\omega_v = \sqrt{(\Sigma K)/m} \quad (37)$$

General Case: Y-Axis Symmetrical. If the mounts are placed symmetrically with respect to the y axis; the last two terms in Equations (28), (30) and (32), and the last term in Equations (29) and (31) are zero. This then gives two sets of three simultaneous equations that can be solved for the

Table I—Natural Frequencies in Cycles per Second for Bottom Mounts in Horizontal Plane

Plane and Mode		Calculated from: Computed radii of gyration	Calculated from: Experimental radii of gyration	Test results
xz	ω_H	53.0	46.6	47.8
xz	ω_L	7.8	6.8	6.7
yz	ω_H	47.1	42.7	44.1
yz	ω_L	13.0	12.2	11.8
xy	ω_T	37.8	34.6	33.4
	ω_v	19.0	19.0	19.2

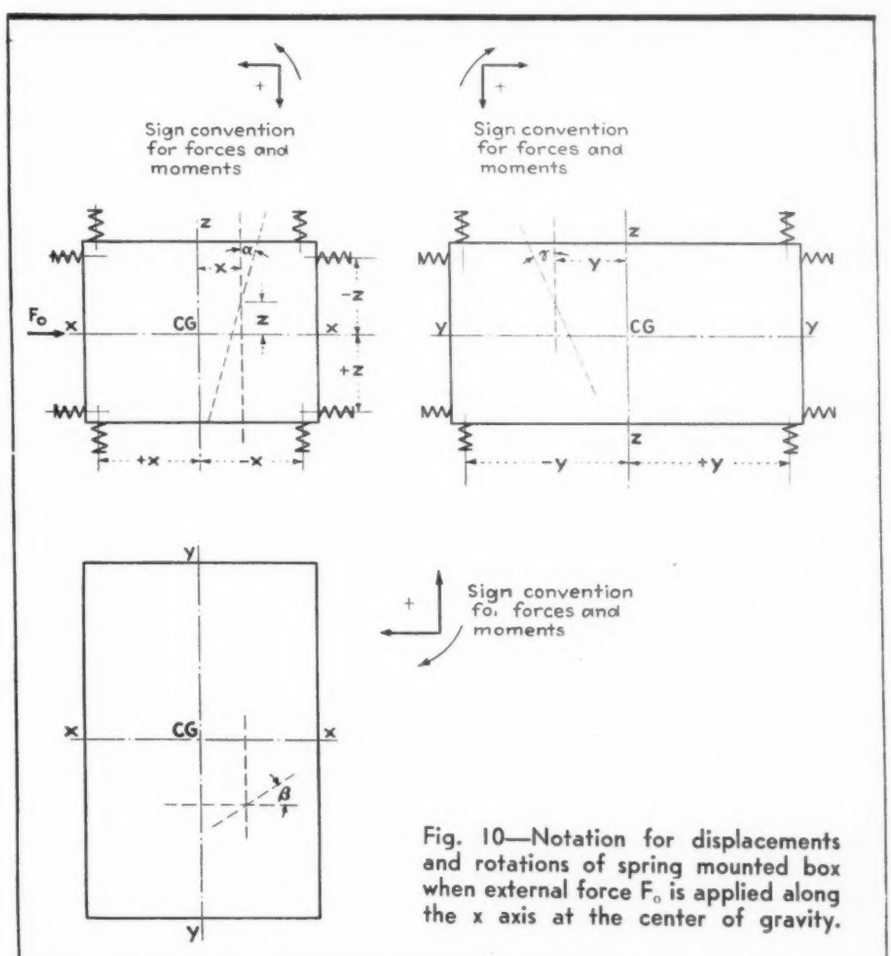

Fig. 10—Notation for displacements and rotations of spring mounted box when external force F_o is applied along the x axis at the center of gravity.

natural frequencies. Proceeding as before, two third-order determinants are set up and equated to zero to obtain natural frequencies.

Where

$$C_1 = mI_yI_z$$

$$C_2 = I_yI_z \Sigma C_z + m I_y (\Sigma C_z X^2 + \Sigma C_z Y^2) + I_z (\Sigma K X^2 + \Sigma C_z Z^2)$$

$$C_3 = \Sigma C_z I_y (\Sigma C_y X^2 + \Sigma C_z Y^2) + I_z (\Sigma K X^2 + \Sigma C_z Z^2) + m (\Sigma C_y X^2 + \Sigma C_z Y^2) (\Sigma K X^2 + \Sigma C_z Z^2) - m (\Sigma C_z YZ)^2 - I_z (\Sigma C_z Z)^2 - I_y (\Sigma C_z Y)^2$$

$$C_4 = \Sigma C_z (\Sigma K X^2 + \Sigma C_z Z^2) (\Sigma C_y X^2 + \Sigma C_z Y^2) + 2 (\Sigma C_z Z) (\Sigma C_z Y) \times (\Sigma C_z YZ) - \Sigma C_z (\Sigma C_z YZ)^2 - (\Sigma C_z Z)^2 (\Sigma C_y X^2 + \Sigma C_z Y^2) - (\Sigma C_z Y)^2 (\Sigma K X^2 + \Sigma C_z Z^2)$$

$$C_5 = m^2 I_z$$

$$C_6 = m^2 (\Sigma C_y Z^2 + \Sigma K Y^2) + m I_z (\Sigma C_y + \Sigma K)$$

$$C_7 = m (\Sigma C_y + \Sigma K) (\Sigma C_y Z^2 + \Sigma K Y^2) - I_z \Sigma C_y \Sigma K - m (\Sigma K Y)^2 - m (\Sigma C_y Z)^2$$

$$C_8 = \Sigma C_y \Sigma K (\Sigma C_z Z^2 + \Sigma K Y^2) - \Sigma C_y (\Sigma K Y)^2 - \Sigma K (\Sigma C_y Z)^2$$

The two equations that result are

$$C_1\omega^6 - C_2\omega^4 + C_3\omega^2 - C_4 = 0 \qquad (38)$$

and

$$C_5\omega^6 - C_6\omega^4 + C_7\omega^2 - C_8 = 0 \qquad (39)$$

APPLICATION OF APPROXIMATE EQUATION IN GENERAL CASE. For the completely general case, where the mounts are not symmetrical about the y axis, the general solution of the six equations for forces and moments is tedious. If it is necessary to obtain a solution, it is advisable to substitute numerical values in the force and moment equations.

To check the accuracy of the approximate equations derived in this article, a box mounted on spring bottom mounts was fastened to a vibrating table and tested.

A comparison of the natural frequencies in cycles per second of the test setup shown in Fig. 11, as obtained by calculation and as obtained by test is given in Table I. For this particular wooden test box the difference between radii of gyration computed on the basis of uniform mass and those found from test is rather larger than might be expected from average apparatus. Despite this difference, the frequencies obtained by using this assumption of uniform mass are reasonably close to the test values.

To check Equations (38) and (39), the model box with back top mounts shown in Fig. 12 was tested. Table II gives a comparison of the results of this test with frequencies computed by the exact method, Equations (38) and (39), and by the approximate method, Equations (34) to (36). While the average error of the approximate method is somewhat larger than for the exact method, the approximate method gives results that are sufficiently accurate for many purposes.

CONCLUSION. The charts presented in Part I definitely point out the desirability of mounts that have a horizontal-to-vertical spring constant ratio of less than 0.5. If this ratio C/K is smaller than 0.5, it is generally feasible to arrange the mounts so that all frequencies are nearly equal to or less than the vertical frequency. With large C/K ratios, the higher rocking frequencies and the torsional frequency will be higher than the vertical frequency, and the useful isolation range is reduced.

Where it is not feasible to use bottom mounts only, it is recommended that the mounts be symmetrically placed about the vertical axis if possible, as this will facilitate the calculation of natural frequencies.

Table II—Natural Frequencies in Cycles per Second for Mounts Symmetrical About Y—Axis

$\omega^2/(K/m)$	Exact method	Approx. method	Test
4.80	20.8	23.3	21.4
7.23	25.6	30.6	25.2
11.23	31.9	32.4	28.2
21.26	43.9	42.3	40.9
22.40	45.2	44.3	42.5
26.77	49.3	47.4	53.0

CAUTION. The assumptions underlying this discussion should be rather carefully kept in mind if agreement between theory and practice is to be maintained.

1. The foundation is infinitely rigid. This is definitely not true. When the apparatus weighs several hundred pounds, it is difficult to find a deck that approaches this condition.

2. The apparatus is rigid. This assumption is generally good for small boxes, but vibration of parts may easily occur in heavy apparatus. If the part is a fair percentage of the total weight, it may change the natural frequencies.

3. The spring constants will not change with time. All materials will drift somewhat with time. Consequently, some change may occur in their characteristics. It is probably wise to test mounts under load over a period of time, especially if the resilient element is not of metal.

4. The mounts are weightless. This, of course, is not true. At high frequencies, resilient material may transmit sound.

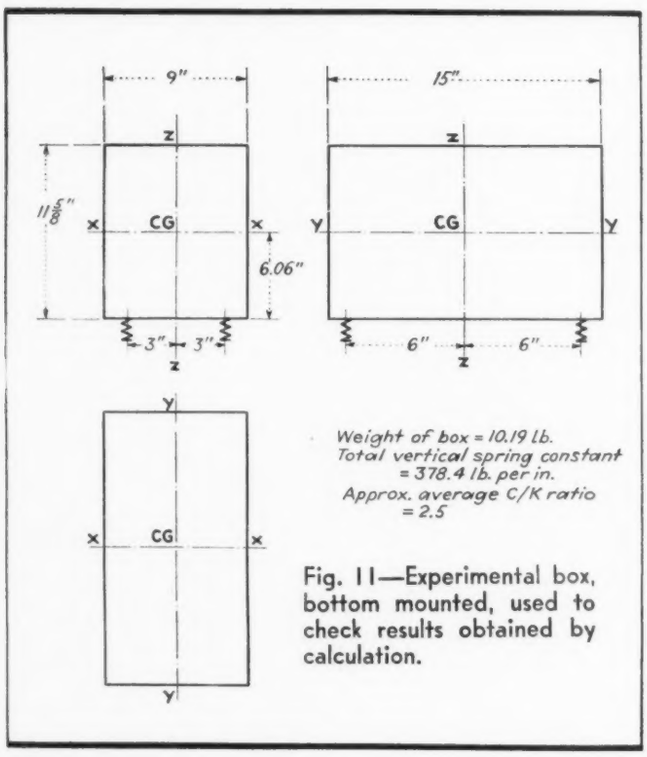

Fig. 11—Experimental box, bottom mounted, used to check results obtained by calculation.

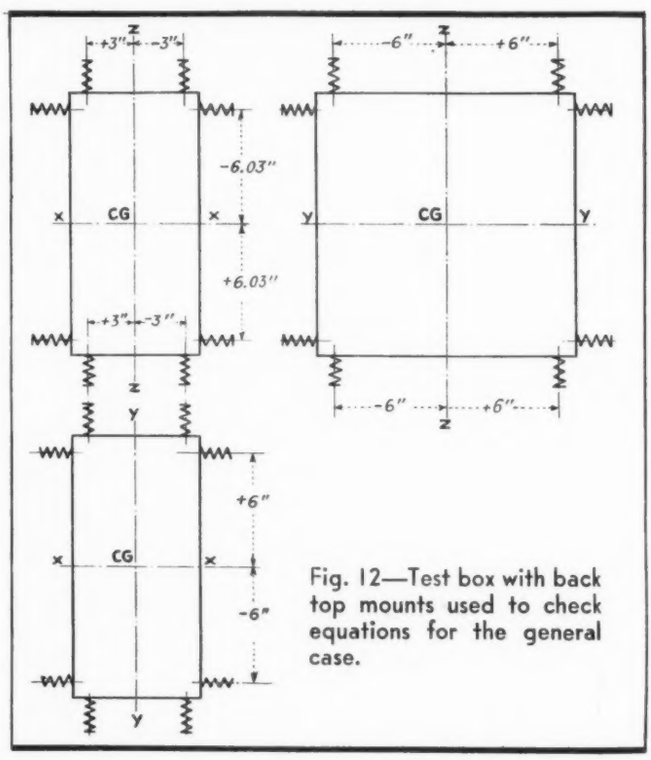

Fig. 12—Test box with back top mounts used to check equations for the general case.

NEW BOOKS

Recommended Practices for Automotive Flash-Butt Welding

Prepared and published by the American Welding Society, 33 West 39th Street, New York 18, New York. 22 pages, 6 x 9 in., gray paper covers. Price $0.30.

Prepared by the AWS Automotive Welding Committee and based on a survey of flash-butt welding as used by the automobile industry, this booklet is an adaptation of the fundamental data on flash-butt welding originally published in the Society's "Recommended Practices for Resistance Welding." Included are sections on automotive-type steels; flash-welding equipment; design considerations; tooling; and inspection. Typical automotive flash-welding assemblies showing both good and poor joint designs are illustrated by sketches taken from actual production drawings.

Electronics in Industry

GEORGE M. CHUTE. *461 pages, 5½ x 8¼ in., black clothboard covers. Published by the McGraw-Hill Book Co., Inc., 330 W. 42nd St., New York 18, N. Y. Price $5.00.*

A detailed description and explanation of the various types of electronic equipment used in industry. Circuits are traced in terms of electron flow instead of the conventional current flow, and new standardized symbols are followed. Examples of increasing complexity show how tubes and tube-operated circuits work. The material is not arranged according to the conventional textbook pattern but is aimed to maintain interest. The book is based on an evening study course in industrial electronics taught by the author and sponsored by the University of Michigan Extension Service.

The Phase Rule and Its Applications

ALEXANDER FINDLAY. *313 pages, 5½ x 8¾ in., black clothboard covers. Published by Dover Publications, 1780 Broadway, New York 19, N. Y. Price $2.75.*

This famous exposition of the Phase Rule and its applications is entirely non-mathematical in character, the author's aim being to explain as clearly as possible the principles underlying the phase rule, and to illustrate their application to the classification and investigation of equilibria by means of a number of cases studied in detail. This is the revised eighth edition of the book.

Polar Molecules

P. DEBYE. *172 pages, 5½ x 8¾ in., black clothboard covers. Published by Dover Publications, 1780 Broadway, New York 19, N. Y. Price $3.50.*

This famous book contains a survey of questions that arise in connection with the conception of polar molecules as systems characterized by a permanent electric moment. The absolute measurement of polarity based on this idea can be used to determine the geometrical arrangement of the atoms in a molecule. The introduction of quantum theory makes it possible to develop in detail the connection between polarity and the phenomena of dispersion and absorption, especially in the infra-red.

Introduction to X-Ray Metallurgy

A. TAYLOR. *400 pages, 6¼ x 9¾ in., red clothboard covers. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York, N. Y. Price $7.50.*

A text of the physics of X-rays and their application to metallurgy. Data on thermal equilibrium, internal stress and strain, crystal texture and phase identification are included. The choice of an X-ray tube is discussed together with the identification of the spectra on the powder photograph, the calculation of interatomic distances, the crystal structures of the metals, and the examination of ternary alloy systems by X-ray. Applications of X-rays to depth of cold working, fiber structure, radiography of welds, emission spectra and absorption edges are described.

Notebook on Machining Stainless Steels

Prepared and published by The Carpenter Steel Co., Reading, Pa. 116 pages, 6 x 8¾ in. Personal copies obtained gratis through Carpenter Steel Co. representative. Additional copies available. Price $0.50.

A notebook of useful information designed to simplify stainless machining and provide longer tool life, higher speeds, and better finishes. Each chapter contains a check chart that lists potential "trouble spots" together with remedial information. Detailed data on turning, drilling, tapping, threading, milling, and broaching is included. A chapter on lubrication discusses proper lubricants and coolants for various types of jobs. The handbook is illustrated throughout. Typical stainless parts are shown and blueprints are provided for specific jobs. Tables indicate recommended speeds and feeds for machining different types of stainless steel.

Rockets

ROBERT H. GODDARD, *117 pages, 6 x 9½ in., black clothboard covers. Published by the American Rocket Society, 29 W. 39th Street, New York, N. Y. Price $3.50.*

The untimely death of Dr. Goddard cut short the work of the world pioneer in rocketry and jet propulsion. At his death, both of Dr. Goddard's famous technical reports, originally published by the Smithsonian Institution in 1919 and 1936, were out of print. These reports, entitled "A Method of Reaching Extreme Altitudes" and "Liquid —Propellant Rocket Development," are now made available in book form. The data in these reports are so fundamental that they are considered the foundation for all present-day developments in jet propulsion and rocketry.

Fundamentals of Alternating Current Machines

A. PEN-TUNG SAH. *466 pages, 6 x 9 in., dark green clothboard covers. Published by the McGraw-Hill Book Co., Inc., 330 West 42nd St., New York 18, N. Y. Price $5.00.*

Although primarily written from the viewpoint of the operating engineer rather than that of the designer, this text is notable for its direct and concise explanations. Particular stress is laid on the operating problems of the machines. Design formulas are generally avoided, but the experimental determination of the machine parameters is fully discussed after the theory is explained. To facilitate explanation, the theory is developed from the standpoint of circuits.

Involutometry and Trigonometry

WERNER F. VOGEL, *Associate Professor of Engineering, Wayne University. 321 pages, 7¾ x 10¾ in., black and red board cover. Published by Michigan Tool Co., 7171 East McNichols Road, Detroit 12, Michigan. Price $20.*

In this volume is presented, for the first time, new data of much practical value to the gear engineer. The subject matter includes a rationalization of the new science of involutometry and its relation to other branches of mathematics.

The preparation of this volume is in keeping with the policy of Michigan Tool Company of pioneering the development of basic information and short-cuts for the gear engineer.

This book contains a Main Table and three Appendices. The Main Table, 182 pages, is a complete listing of trigonometric and involutometric functions. The trigonometrical table is more complete and more conveniently arranged than most tables of functions. Values of trigonometric functions, radians, and involute functions are given in a double page arrangement, so that the user can read any function of an angle when any other function is known without the necessity of finding the angle itself. The tables as presented simplify engineering calculations related to the design and manufacture of products based on involute curves and surfaces, such as gears, gear tools and splines.

Under the heading "Miscellaneous Mathematical Tables" are conversion tables, formulas for involutometry, cycloid tracing, involute tracing, and polygon tables.

Tables of angular pitch and chordal pitch; formulas for spur, helical and spiral gears; tables of tooth parts; and data for involute gears are presented in an appendix.

To simplify the calculation of the involute, the analytical foundations underlying the calculations are made clear by a discussion of the geometrical properties of this curve and of related curves. This discussion, presented in the third appendix, is a concise classification and introduction to involutometry.

The author has made use of numerous excellent line drawings and charts.

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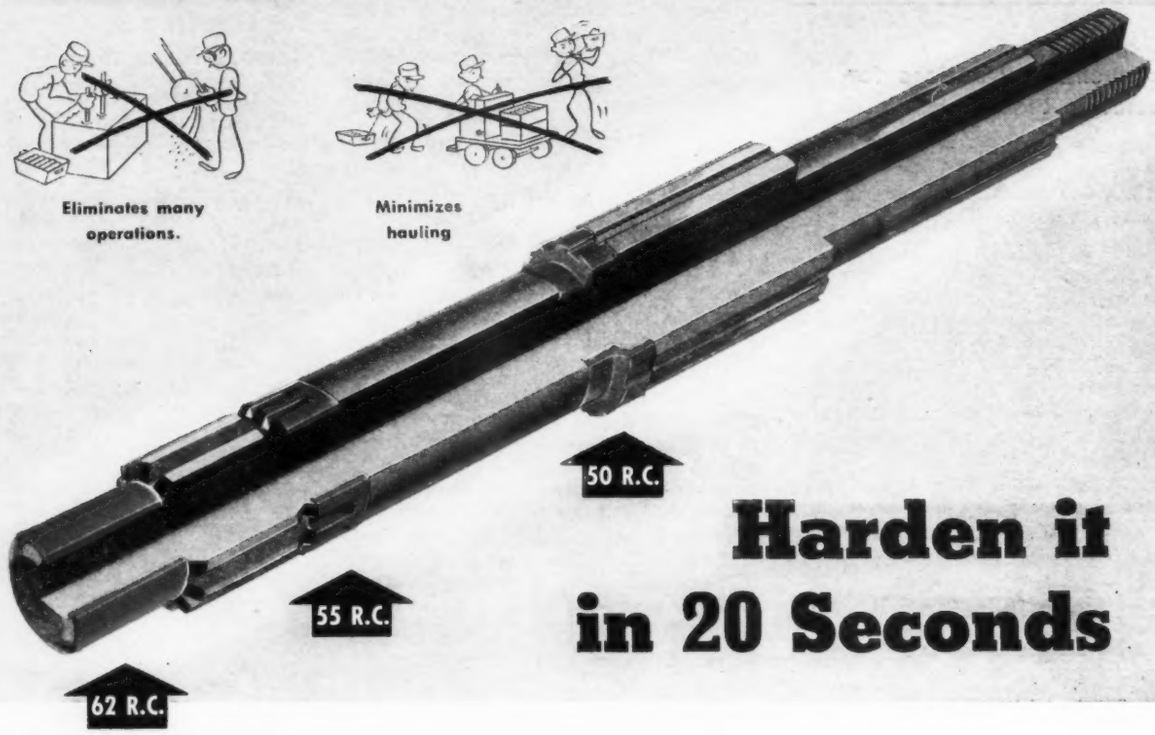

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Formulas for Determining the Properties Of Copper Wire in the American Wire Gage

CARL P. NACHOD, *Nachod & U. S. Signal Company*

THE ACCOMPANYING table gives simple formulas of sufficiently close accuracy to obtain the properties of any size copper wire in the American or Brown and Sharpe Gage. With these formulas of the wire properties in terms of circular mils and a few powers of 2 memorized, the entire copper wire table in the American Wire Gage may be reproduced at will with an error rarely exceeding 3 percent.

The following example is solved by three methods given, and the values obtained are compared with those from standard wire tables.

EXAMPLE 1: What is the resistance of 1,000 ft. of No. 17 wire?

SOLUTION: (First Method) Let n = 18 and use Equation (1), thus

$100 \times 2^{10-\frac{18}{3}} = 1{,}600\ C_m$ for No. 18 wire

Using Equation (1a),

$\frac{1600}{0.79} = 2{,}020\ C_m$ for No. 17 wire. Table 2,048 C_m

From Equation (3),

$\frac{10^4}{2020} = 4.95$ ohms per 1,000 ft. Table 5.06 ohms

SOLUTION: (Second Method) Let $n = 20$ and use Equation (11), thus

$10^{10-\frac{20}{10}} = 10$ ohms per 1,000 ft.

Since No. 17 wire has one-half the resistance of No. 20 wire due to the fact that C_m is doubled for each three changes of larger gage, the resistance of 1,000 ft. of No. 17 wire is therefore approximately 5 ohms. Table—5.06 ohms

SOLUTION: (Third Method) From Equation (8),

$C_m = 2^{\frac{50-17}{3}} = 2^{11} = 2{,}048$ Table 2,048 C_m

Thus, from Equation (3),

$\frac{10^4}{2{,}048} = 4.88$ ohms for 1,000 ft. Table 5.06 ohms

C_m = Circular Mils

(1) $C_m = 100 \times 2^{\frac{10-n}{3}}$ (for gages that are multiples of 3)

Gage No.	4/0	3/0	2/0	0	1	2	etc;
n	−3	−2	−1	0	1	2	etc;

(1a) Multiply or divide by 0.79 for C_m of next higher or lower gage number if necessary to make n fit (1) except when using formulas (7) and (8) below.

(2) $d_m = \sqrt{C_m}$

(3) Ohms per M ft. at 20 deg. $C = \frac{10^4}{C_m}$

(4) Ft. per ohm $= \frac{C_m}{10}$

(5) Lb. per M ft. $= \frac{3.1 \times C_m}{10^3}$

(6) Ft. per lb. $= \frac{3.1 \times 10^5}{C_m}$

(7) $C_m = 10 \times 2^{\frac{40-n}{3}}$ (When n must be one more than a multiple of 3)

(8) $C_m = 2^{\frac{50-n}{3}}$ (When n must be one less than a multiple of 3)

For gages in multiples of 10, the following equations apply:

(9) $C_m = 10^{5-n/10}$

(10) $d_m = 10^{\frac{50-n}{20}}$

(11) Ohms per M ft. $= 10^{n/10-1}$

(12) Ft. per ohm $= 10^{4-n/10}$

(13) Lb. per M ft. $= 3.1 \times 10^{2-n/10}$

(14) Ft. per lb. $= 3.1 \times 10^{n/10}$

Some powers of 2:

$2^3 = 8$ $2^9 = 512$

$2^6 = 64$ $2^{12} = 4{,}096$

COPPER WIRE FORMULAS FOR AMERICAN WIRE GAGE

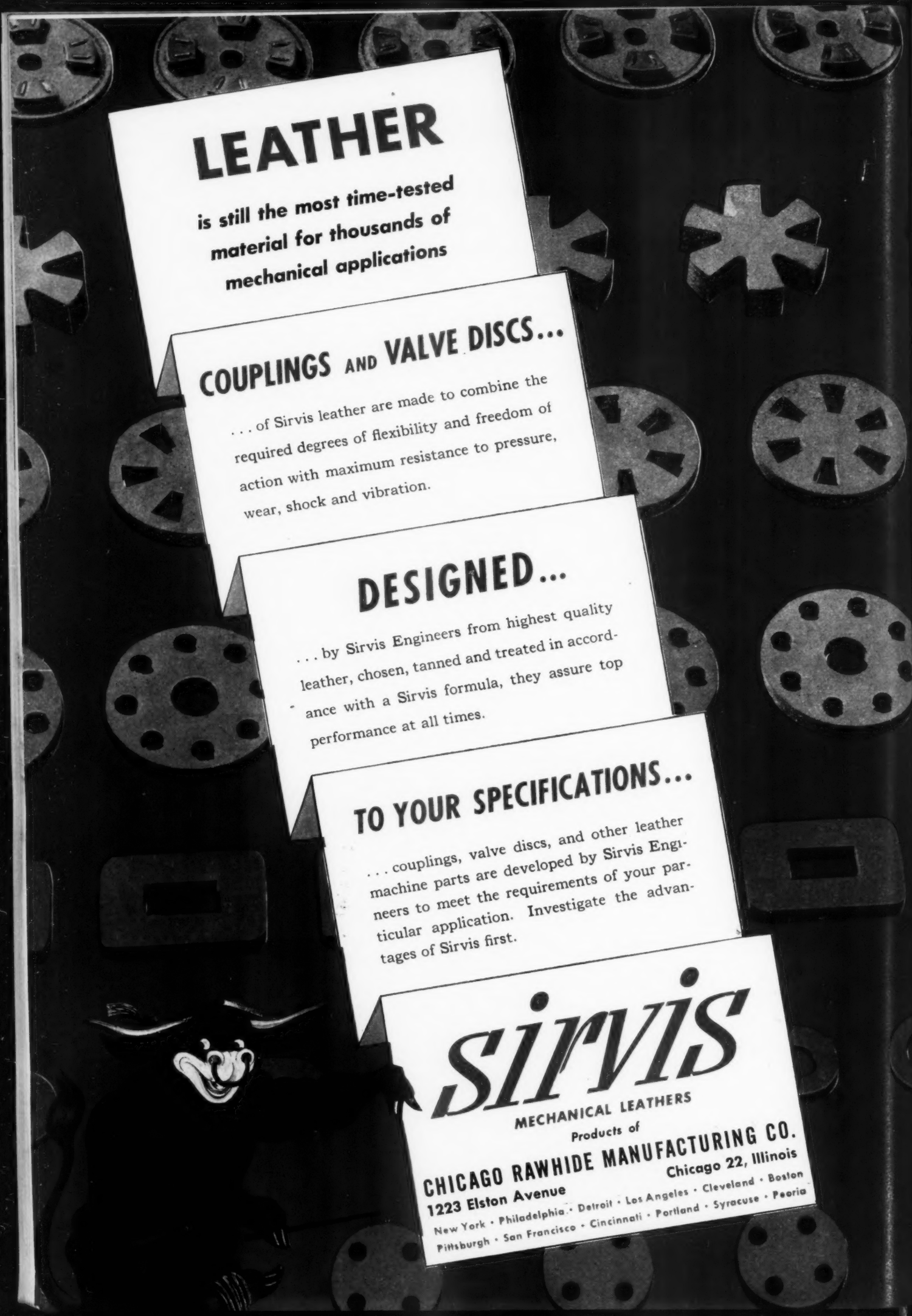
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Chart for Determining the Properties Of Copper Wire in the American Wire Gage

CARL P. NACHOD, *Nachod & U. S. Signal Company*

THE ACCOMPANYING chart of the properties of copper wire in the American or Brown and Sharpe Gage is plotted as four straight lines for six properties. In order to keep two of these within the chart, graphs for ohms per 100,000 ft. and lb. per 100,000 ft. are drawn instead of per 1,000 ft. as usual. Hence, each of the scale readings for these must be divided by 100 in order to obtain the standard values. Two properties, ohms per lb. and lb. per ohm are omitted as having far too great a range.

The American or Brown and Sharpe Gage is a geometrical series having 4/0 wire of 460 mils for the first term in the series, and No. 36 wire of 5 mils dia. for the 39th term, and extended to No. 40. When 4/0, 3/0, 2/0, and 0 gage numbers are taken as −3, −2, −1 and 0 respectively, the equation for the diameter in mils of any wire is: $d_m = 460r^{n+3}$

in which 1/r is found to be 1.1229.

Circular mils are obtained from the formula

$$C_m = 460^2 r^{2(n+3)}$$

in which $\frac{1}{r^2}$ is found to be 1.261 or $r^2 = 0.7931$.

Therefore: $C_m = 211{,}600 \times 0.7931^{n+3}$

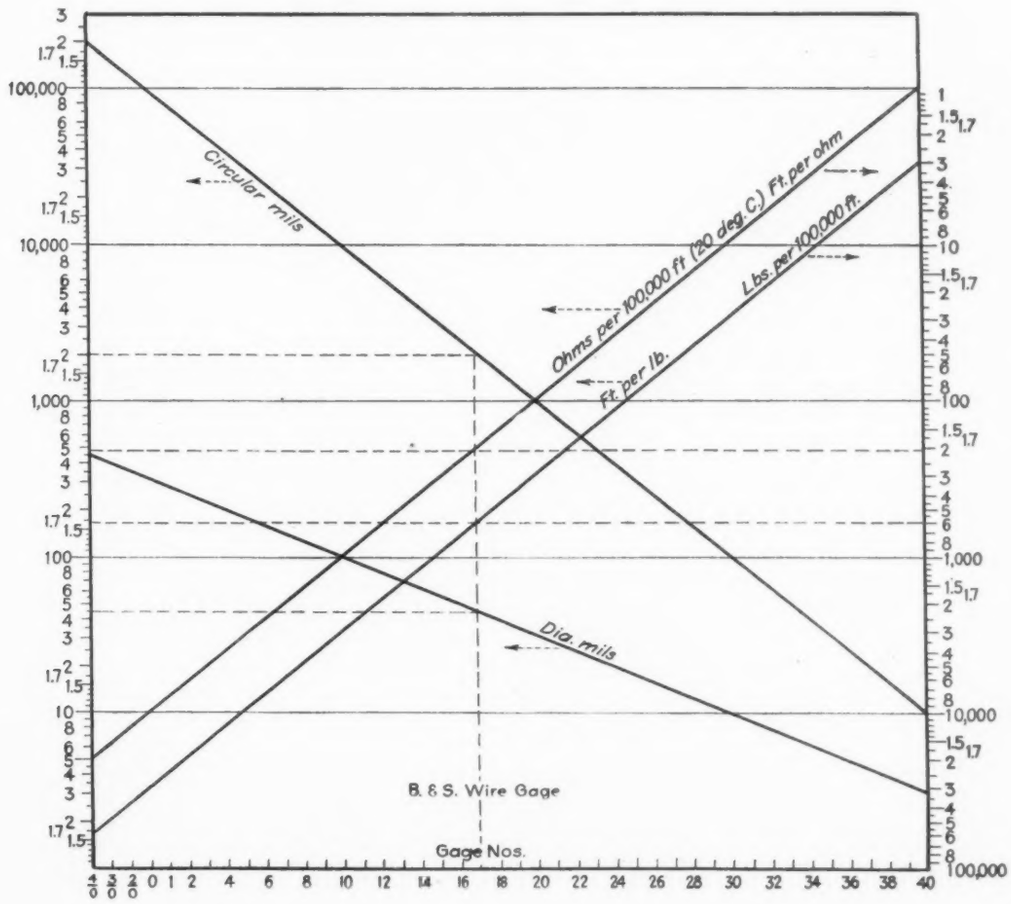
This is the American Gage formula for circular mils.

The following example is solved on the chart.

Example: What is the weight per 1,000 ft., ft. per ohm, ohms per 1,000 ft., ft. per lb., circular mils, and dia. mils of No. 17 gage copper wire.

Solution: Erect a perpendicular from the abscissa at the proper gage number, in this case, No. 17. At the points of intersection of this perpendicular with the four plotted lines, run across horizontally to the ordinates as indicated by the dashed lines on the chart. The required values obtained for this gage are therefore:

dia. mils	= 45.3
ft. per lb.	= 161
lb. per 100,000 ft.	$= \frac{620}{100}$ = 6.2 lb. per 1,000 ft.
ohms per 100,000 ft.	$= \frac{506}{100}$ = 5.06 ohms per 1,000 ft.
ft. per ohm	= 197
circular mils	= 2,048



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Special-Purpose Sheet Steels • Stainless Steel Sheets, Bars and Wire

NEW MATERIALS AND PARTS

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Capacitors

Aerovox Corp., New Bedford, Mass.

Maximum watt-second energy and minimum weight are claimed for these capacitors by ratings consistent with the duty cycle. Each unit is provided with internal connections to minimize inductance. To handle high currents, units are divided into sections. This reduces the currents in individual connections. Units cover watt-second ratings from 50 to 540, in voltages from 2,000 to 4,000 inclusive.

Fan-Cooled Motor

General Electric Co., Schenectady, N. Y.

Induction motor available in standard, explosion-proof, and dust-explosion-proof types from 1 to 1,000 h.p. and for use in Class I Groups C and D and Class II Groups E, F, and G hazardous locations. Sealed end shields, inside joints, and a one-piece, double-shell stator protect windings, punchings and rotating parts from moisture or corrosive elements. Close-running shaft fits, supplemented by a rotating seal, are claimed to keep dirt from entering the motor along the shaft. Punchings and windings within the inner shell of the motor are cooled by a non-sparking external fan that is protected by a cast-iron housing with a screened air-intake opening. The cast iron bearing housings are cast integral with the end shields as a barrier to the entrance of foreign materials. An inner cast-iron bearing cap makes a complete inclosure for the bearing.

Direct-Coupled Amplifier

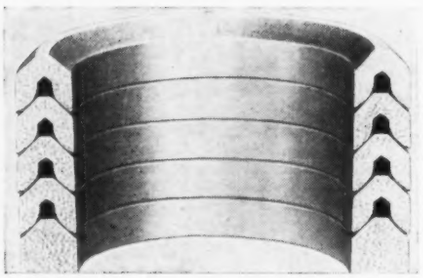
Amplifier Co. of America, 398 Broadway, New York 13, N. Y.

Developed for amplifier applications requiring a wide pass-band, low inherent amplitude, and cross-modulation distortion, this unit utilizes a signal self-balancing and current drift-correcting direct-coupled output circuit. Response is 20 to 20,000 cycles ± 1 db. It is claimed to develop 23 watts with less than 1 percent total distortion. Overall gain is 96 db. Two independent inputs (each of 500,000 ohms) are provided. Balanced output terminals are provided for 4, 8, 16 and 500 ohms. In-between terminals provide additional output impedances of 1, 2, 6, 10, 12, 83, 100, 125, 150, 166 and 175 ohms. Power consumption is 150 watts. Overall dimensions 17½ × 10 × 10 in. Net weight 40 lb. Unit is furnished with input cables.

Packing Rings

Greene, Tweed & Co., Bronx Blvd., 238th St., New York 66, N. Y.

Known as Palmetto Pyramid packing, this product is available in molded ring form to exact dimensions. The cutaway illustration of assembled set shows the method of

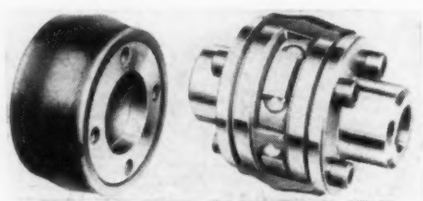


assembly inside the stuffing box. The numerically keyed cross-sectional view shows how the assembled packing functions. The bottom ring is installed first and receives the full impact of the pressure on the power stroke. This ring expands the wedge-shaped lip of each ring above it so that the packing is pressed against both the rod and side walls. Binding is prevented by infiltration of fluid between the rings and storage of fluid in the reservoir. Sets can be supplied to withstand oil, water, and steam at any pressure, and temperatures to 600 deg. F.

Flexible Couplings

The Gear Grinding Machine Co., 3901 Christopher St., Detroit 11, Mich.

Ballflex, an angular and axial flexible coupling obtains its flexibility mechanically. Friction is reduced by using hardened and ground steel balls through which all torque is transmitted. Free rolling movement in



any direction eliminates side thrust and the bending or springing of parts, both in the coupling and the mounting bearing. Ballflex accommodates parallel misalignment to 1/8 in., and angularity to 3 deg. No lubrication is required, as the unit is packed with lubricant when manufactured and sealed from dust or liquid.

Plastic Material

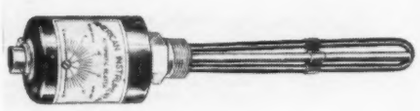
Duorite Plastic Industries, 8564 W. Washington Blvd., Culver City, Calif.

Duroflex is a non-vulcanizable thermoplastic, and can be reclaimed by melting innumerable times without additives. One-piece Duroflex molds can be used to produce articles with all types of back drafts and undercuts, because they can be stretched when it is necessary to remove castings. Casting surfaces require no lacquers or wax parting agents, as Duroflex is a non-porous material with a texture that duplicates the contours of a model or pattern. Duroflex molds are claimed to withstand temperatures to 225 deg. F. Duroflex can also be used as a strip-coating material and as a molding compound for many soft rubber-like articles.

Immersion Heater

The American Instrument Company, Silver Spring, Md.

An automatic electric safety immersion heater for liquids. A built-in safety control limits the temperature rise of the heater and protects it against overheating and burning out if the liquid level falls and exposes the heating element to air. The heater can be screwed into the walls of a tank or drum through a 1-in. pipe fitting or reducer, and two wires connected from the current supply to the heater. Two or more of these automatic heaters can be installed on one application and adjusted for step tempera-



ture control, or, where it is desirable to install more than one heater in the same vessel, the Aminco Safety Heater can be used to regulate one or more non-automatic heaters through a relay. By turning an indicating screw, the heater can be set as desired from room temperature to 350 deg. F. The heater will maintain this temperature within plus or minus 50 deg. F. The sheath of this heater cannot become electrically energized, thus there is no danger of the material becoming electrically charged. A pilot light indicates when the heater is functioning. These heaters are available with copper sheaths for heating water, or with steel sheaths for heating light oils, in ratings of 250, 500, 750, and 1000 watts.

Marking Inks

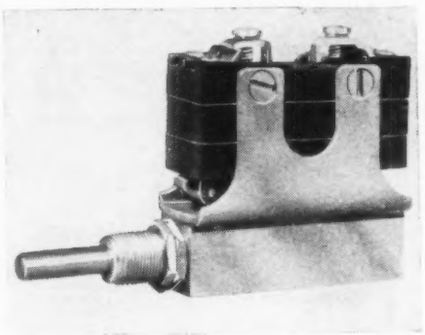
The Acromark Co., 382 Morrell St., Elizabeth 4, N. J.

Inks for trademarking and imprinting plastics, rubber and insulated wire. The ink for rubber marking can be furnished in standard colors in either liquid or paste form. The liquid inks are applied by hand or machine, in standard liquid-type ink fountains and are furnished in drying characteristics to suit the application. The paste form inks are applied by hand, or mechanically in standard ink feeds.

Overtravel Switching Device

Mu-Switch Corp., Canton, Mass.

Designed to provide flexibility in applications where a switch may be subjected to shock and vibration, the type EP-203 Mu-Switch assembly comprises a standard Mu-Switch

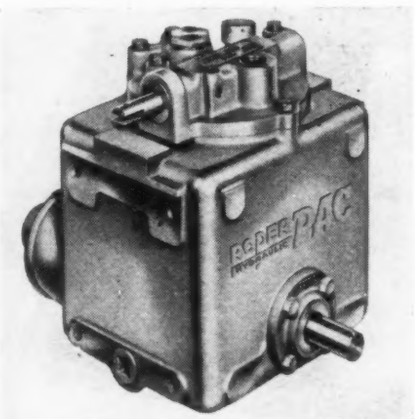


in a rigid bracket in which is incorporated a long-throw overtravel mechanism. The operating force is applied in a direction parallel to the moving elements of the switch. The force and movement characteristics are governed by the over-travel device and are independent of the normal characteristics of the switch assembled to it. In the unit illustrated, the normal overtravel is 9/16 in. and the movement differential at the actuating point is 1/32 in. A threaded shoulder, furnished with double lock-nuts, permits single hole mounting in panels to 1/2 in. in thickness. When assembled with a standard type E Mu-Switch for direct-current operation at loads to one kilowatt, the switch assembly occupies a space of 2 1/4 × 2 1/4 × 7/8 in., exclusive of the threaded shoulder and over-travel plunger. The assembly is furnished with the standard type M or type A Mu-Switch for a.c. loads to 15 amp. or with the type F Mu-Switch for d.c. loads to 1 1/4 kilowatts at 250 volts.

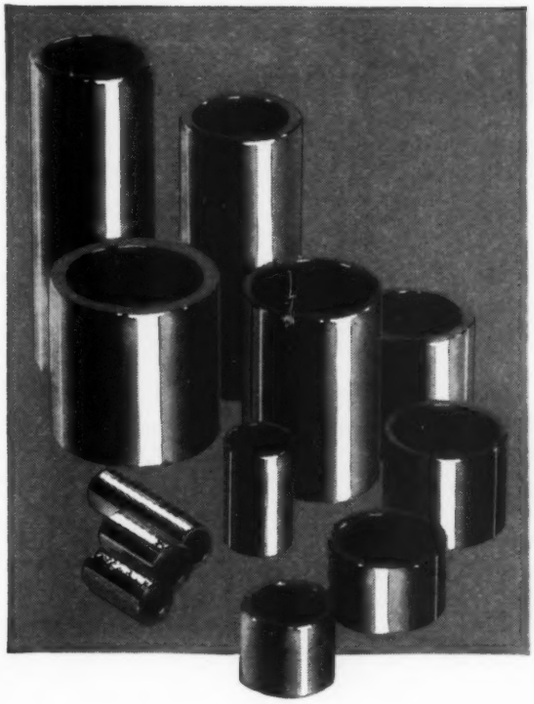
Hydraulic Power Unit

Geo. D. Roper Corp., Rockford, Ill.

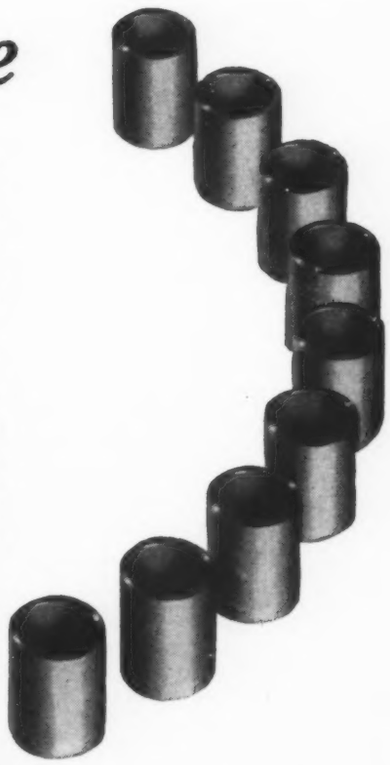
The "Roper-PAC", designed for universal application on farm tractors, contains three fundamental elements, a rotary-gear pump, control valves, and an oil reservoir. The unit



is direct-connected to power take-off, or is chain, gear, or V-belt driven. It is built for either clockwise or counterclockwise rotation and has a range of operating speeds from 500 to 1,000 r.p.m. An unloading valve prevents overloading and overheating. Normal operating pressure is 800 lb. per sq.



Now is the time to Standardize



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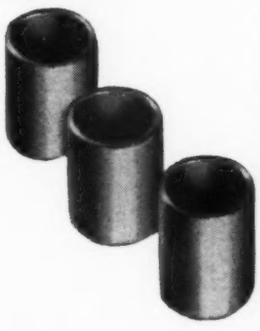

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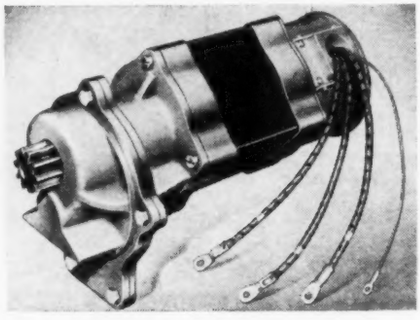
in. and the capacity is 5 gal. per min. at 500 r.p.m. at 600 lb. per sq. in. The sump working capacity is 2 gal. The unit weighs 42 pounds.

Starter-Generator

Electrical Engineering and Mfg. Corp., 4606 Jefferson Blvd., Los Angeles 16, Calif.

Starter and generator are combined in one unit, the same windings serving for both. The starter gear reduction for cranking is 23:1; the generator reduction is 2½:1. Max-



imum starting torque is 160 ft.-lb. Generator output is 500 watts, continuous duty. A torque limiting clutch, built into the unit, prevents damage to the unit from backfire. Coupling to the engine crankshaft eliminates need for belt or chain drives. Adapter mounting brackets are available. Unit measures 9½ in. overall and weighs 17 pounds.

Insulating Varnish

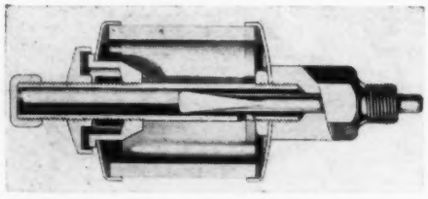
Irvington Varnish and Insulator Co., Irvington, N. J.

Known as Harvel 912C, this phenol-aldehyde varnish is claimed to cut curing time 50 percent. It is also claimed to provide greater dip tank and storage stability, while maintaining favorable electrical and chemical properties.

Vibrating Rod Bottle Oiler

Oil-Rite Corp., 3466 South 13th St., Milwaukee 7, Wis.

This unbreakable plastic unit, automatic in operation, is refilled without removal from its installation. Oil is fed from reservoir through an oilport to a feed rod that slides in a fitted hole in the base and rides on the journal. A slight vibration is caused by

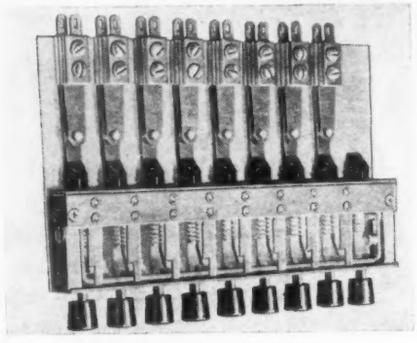


the directional drag of the rotating shaft on the feed rod, resulting in a pumping action. As the bearing becomes warm, heat is conducted through the rod to the reserve oil, causing the air in the reservoir to expand. The pumping and thermal action forces oil down the feed rod to the bearing. Available in standard capacities of 1/2, 1, 2, 4, 8, 16, and 32 ounces, having 1/8, 1/4, 3/8 or 1/2 pipe thread.

Pushbutton Switch

General Control Co., 1200 Soldiers Field Road, Boston 34, Mass.

The "Master" Model MPB, a nine-position push button switch made in both locking and non-locking frame types. The locking frame type has eight positions and one reset

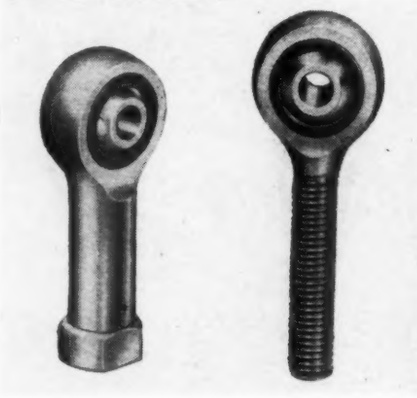


position. Any switching combination that has been set can be released by one operation of the reset button. All the parts are made of non-corrosive materials. The contacts are silver, permanently riveted to the phosphor-bronze contact springs. The rating of the "Master" Model MPB switch is 5-10 amp. 125 volts, 60 cycles a.c. (non-inductive load).

Spherical Rod End Bearings

The Heim Co., Fairfield, Conn.

Units are available with male or female shanks and have bore diameters varying from 0.190 to 0.500 in. + 0.0015 — 0.0005. Outer ring (shank member) is made of carbon steel with bronze bearing inserts. Ball is

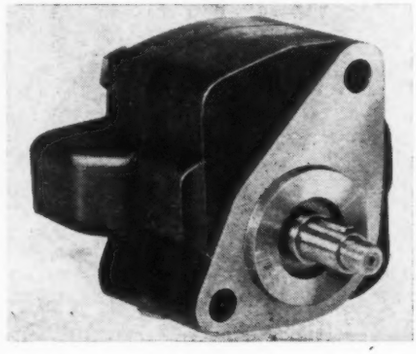


of heat-treated SAE 52100 steel. Bearings are lubricated by an oil groove around the ball that is supplied by a reservoir in the shank. These units have 10 deg. of alignment in either direction.

Hydraulic Pump

The Aro Equipment Corp., Bryan, Ohio.

Gear-type hydraulic pump, Model H657-A, delivers pressure to 2000 lb. per sq. in. with 5½ gal. per min. at 2,800 r.p.m. Volumetric efficiency using S.A.E. No. 10 oil is claimed to be above 90 percent. Overall efficiency above 500 lb. per sq. in. is 85 percent. It



is suitable for hydraulic systems ranging from 100 to 2,000 lb. per sq. in. The pump uses Nitralloy gears in a cast iron body of "sandwich" design, and provides porting to standard pipe sizes. The unit is approximately a 3½ in. cube, with ½ in. dia. x 11/16 in. drive shaft extension, plus threaded portion for 3/8 in. nut.

Silastic-Coated Fibreglas Tape

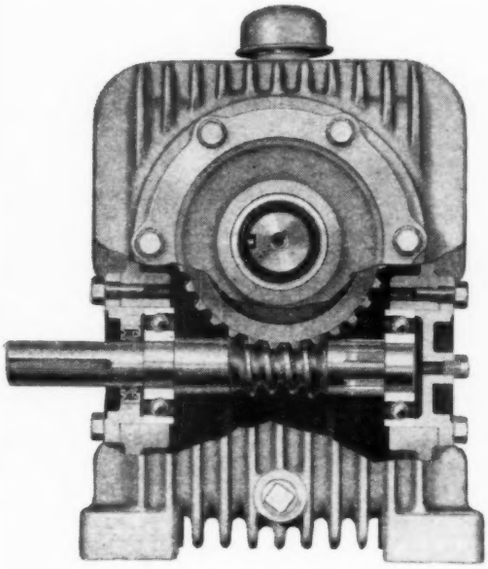
The Connecticut Hard Rubber Company, New Haven, Conn.

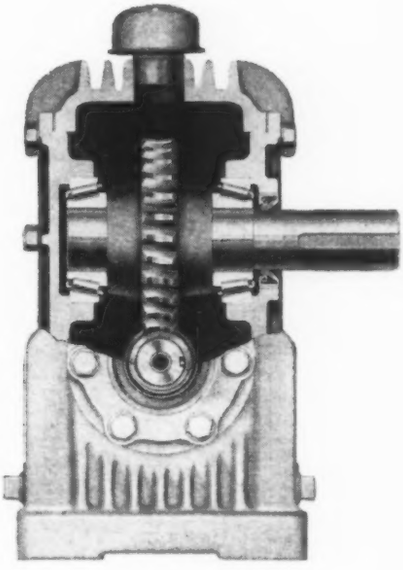
Marketed under the tradename Cohrlastic, this tape is available in rolls of various lengths, widths, and thicknesses. It has high dielectric properties, 1100 volts per mil, and capacity to retain this strength at temperatures to 500 deg. F. and not breakdown and carbonize. It is resistant to arcing and corona cutting, is waterproof and water repellent. Cohrlastic is not sticky and wraps are bound at the ends. Gaskets die-cut from the new material are used in fluorescent transformers and similar places where high dielectric strength is required at elevated temperatures.

Oil Conditioner

Winslow Engineering Company, Oakland, Calif.

Intended for engines from 150 to 250 hp., this conditioner is known as Model No. 8-931-C. It contains eight elements with a total filtering area of 684 sq. in., weighs 35 lb. dry and 65 lb. wet, and has an oil capacity of 3¾ gal. The elements, when





Announcing

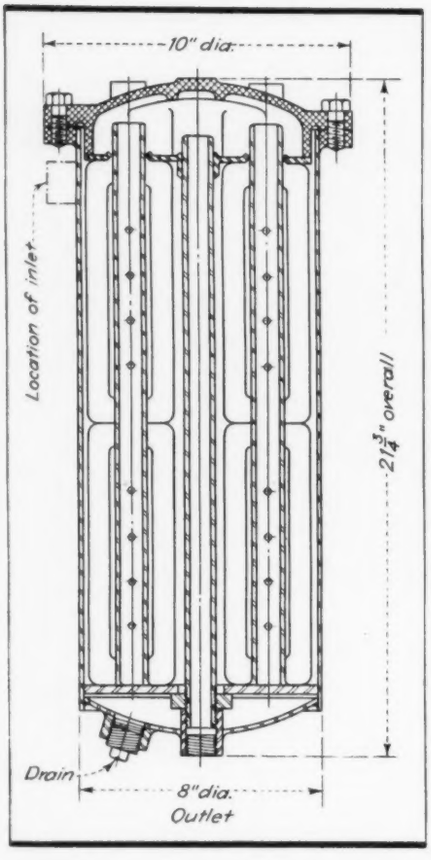
STANDARDIZED CONE-DRIVE SPEED REDUCERS

Smaller center distances, made possible by double-enveloping Cone-Drive gearing with its greater contact area per tooth and more teeth in contact mean remarkable compactness and light weight despite their heavy duty construction. All standard models carried in stock to suit practically every requirement—from fractional hp. to 800 hp. input to the pinion.

Models include—pinion under, pinion over, gear shaft vertical or horizontal. Ratios 5:1 to 70:1. Extended pinion and gearshaft models available. Cooling fins provided on all standard models, greatly increasing thermal capacity. Larger models designed for water cooling where required.

Ask for Bulletin No. 789

CONE-DRIVE DIVISION **MICHIGAN TOOL COMPANY**
7171 E. McNichols Road, Detroit 12, U.S.A.

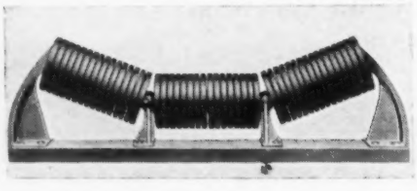


clean, permit a maximum efficient flow of 15 gal. per min. with the oil at 140 deg. F. minimum. The conditioner is so designed, that the oil cannot drain back into the crank-case when the engine is idle. The filter will be full when the engine is started. In the cover assembly, four built-in by-passes, each with a free area equal to the filter outlet, are constructed to prevent dirt from building up on the ball seat and holding the by-passes open.

Troughing Idler

Chain Belt Co., 1600 West Bruce St., Milwaukee 4, Wis.

Known as the Rex Style No. 35, this unit consists of a rubber cylinder vulcanized directly to the assembly tube. The cylinder has multiple grooves molded into it; deep primary grooves for maximum cushioning guard against belt carcass rupture, and shallow secondary grooves provide surface softness for protection against belt cover laceration. For all loading conditions, the cushion idler should be used to support the belt

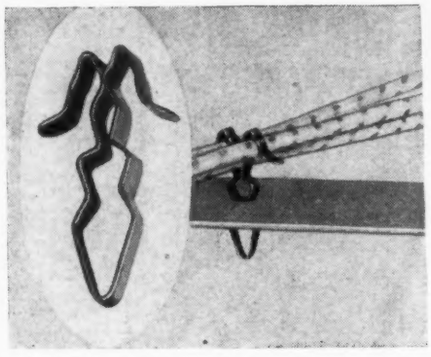


directly under the loading point. Each roll is equipped for high pressure grease lubrication and is furnished with an hydraulic type fitting.

Insulated Wire Clip

Tinnerman Products, Inc., 2041 Fulton Road, Cleveland 13, Ohio.

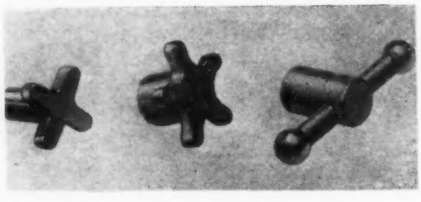
A spring-steel clip to hold two or three parallel wires. This clip is snapped into a ⅜ in. round hole. It locks itself in place ready to receive the wires. No tools are required. It fits in panels from 0.025 to 0.064 in. thick, and holds three No. 16, No. 18, or No. 20 insulated wires.



Hand Knobs

Machine Products Corp., Detroit, Mich.

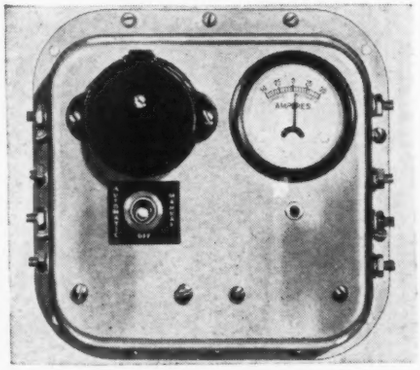
These knobs are cast iron and are supplied either machined or unmachined. The four-star design is available in 6 sizes, the 5-star in 2 sizes, and the "T" handle design in 2 sizes. The hand wheels are available in rough castings or machined, with or without polished steel handles.



Starting Panel

R-B-M Div. Essex Wire Corp., Logansport, Ind.

Designed for gasoline engine-driven refrigeration compressors and pumps, these panels are built with standard automotive magnetic and manual electrical controls. Auxiliary devices such as engine starting solenoids, reverse current contactors and voltage regulators are available for electrical control installations. The panel closes ignition circuit and energizes engine starting solenoid when control device closes, and de-energizes the engine starting solenoid after the engine fires and generator approaches rated voltage.

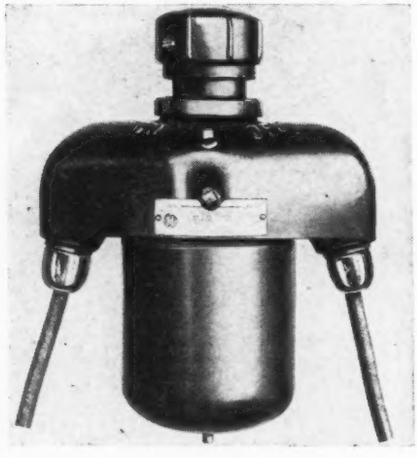


When control device opens, control panel opens ignition circuit, stopping engine. Panel is available with cranking time limiter and cranking interrupter to provide several starts. Hand reset cranking time limiter stops cranking cycle should engine fail to start.

Oil-Fuse Cutouts

General Electric Co., Schenectady 5, N. Y.

Heavy-duty oil-fuse cutouts for overcurrent protection of power distribution systems in the 2,400; 4,160; 4,800 and 7,200 volt classes. These units have universal fuse links



giving higher interrupting ratings, clearing fault currents to 11,000 amp. With this design, 100, 200 and 300 amp. rating cutouts with maximum short circuit interrupting ratings are extended to 5,000 volts. The cutouts are claimed to provide a high degree of safety, as no exposed live parts, and no arc flames are ejected during current interruption.

Industrial Gear-Motor

Electrical Engineering and Mfg. Corp., 4606 W. Jefferson Blvd., Los Angeles, Calif.

A fan-cooled, open-type, two-pole 3,600-r.p.m. motor, with output speed stepped to 8,000 r.p.m. by single-reduction helical gears in a sealed gear box. This motor is claimed to be smaller, lighter, and more

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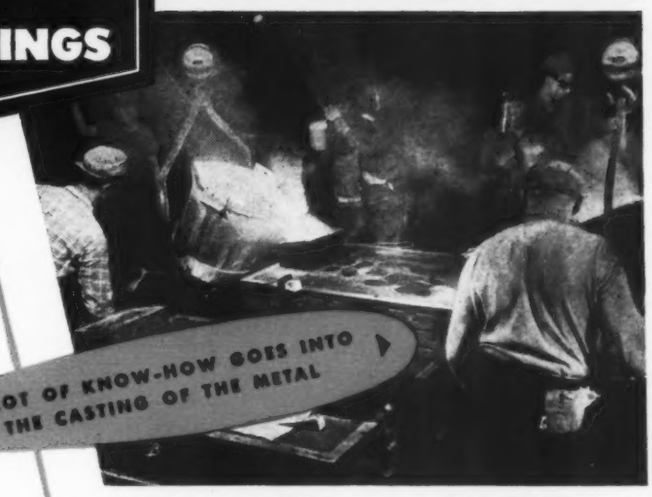

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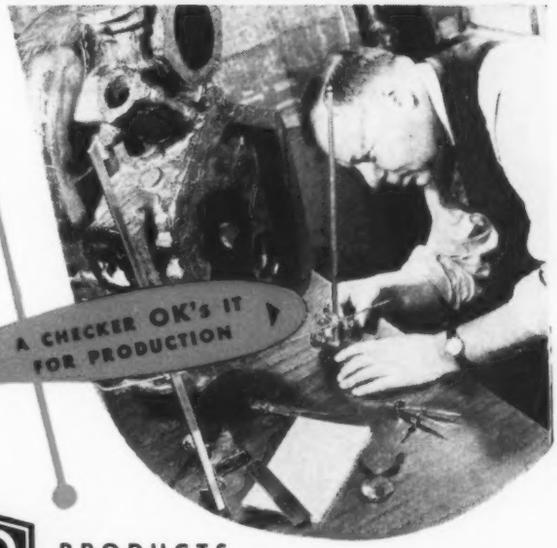

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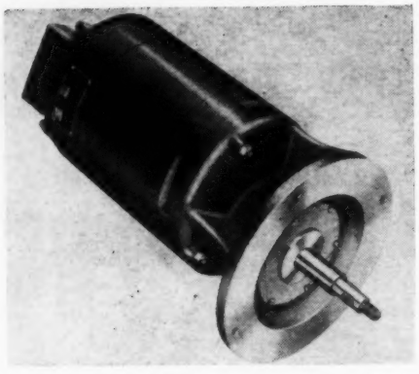


compact than the conventional ungeared motor. The same design, with changed gear specifications, will fit a variety of applications requiring output speeds not ordinarily obtainable with commercial 3-phase, 60 cycle, 220-volt power.

Plastic Film

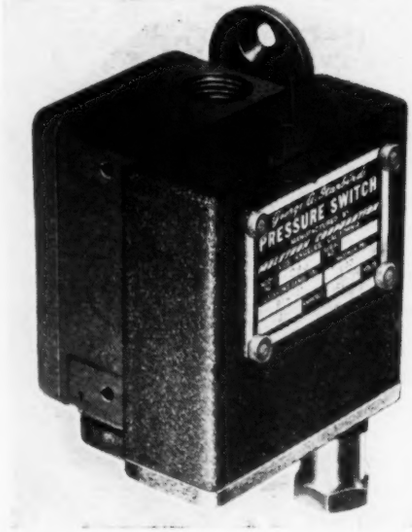
Better Finishes & Coatings Inc., Newark 5, N. J.

"Liquid Envelope" is a peelable protective plastic film for spray application to steel and non-ferrous sheet prior to cutting and fabricating operations. In a film 0.002 in. thick, this material is claimed to protect against scratches, draw marks, die marks and other surface scars incidental to machining and assembly operations. It also serves as a masking material during finishing and for the protection of machined parts in stock and in transit.

Pressure-Operated Switch

Meletron Corp., 950 N. Highland Ave., Los Angeles 38, Calif.

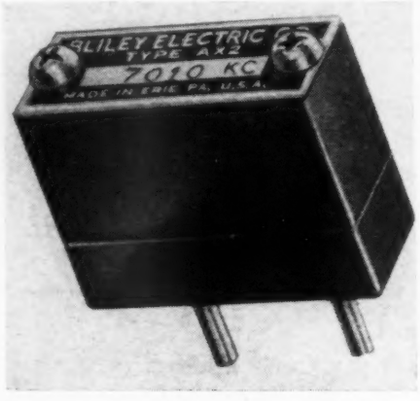
Known as model series 320, this unit has separate housings for pressure and electrical controls for adjustments in the pressure system without disconnecting or exposing the electrical circuit. Various models provide uniform pressure control from 25 to 2500 lb. per sq.in.; minimum differential adjustment 20 lb. per sq.in., maximum 200 lb. per sq.in. Conduit connection: ½-in. Switch is single pole, double throw, 10 amp. at 125 volts a.c. Pressure connection: ¼-in. Weight of entire unit, including mounting bracket, is 3-3/4 lb. Overall dimensions: 3-5/8 × 3-3/4 × 5-1/2 inches.



Plated Crystal

Bliley Electric Co., Erie, Pa.

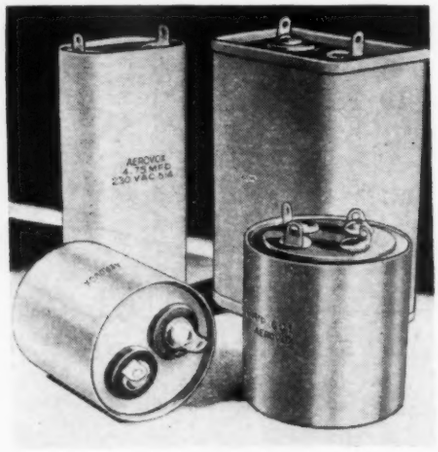
This crystal, Type AX2, has primary electrodes of a micro-thin metal film deposited directly on the major crystal surfaces by evaporation under high vacuum. Secondary electrodes, under spring pressure, clamp the crystal and provide necessary thermal dissipation. This design is claimed to result in better grid current stability over a wide temperature range; improved frequency stability under high drive conditions; and substantial improvement in keying characteristics.



Capacitors

Aerovox Corp., New Bedford, Mass.

Impregnated with Hyvol, a synthetic high-dielectric compound, these capacitors have plated and induction-soldered steel covers. The terminals are treated to eliminate leakage, and the lugs are tinned. Aerovox fluorescent capacitors are available in all standard types, and in special sizes and shapes.



Wax Emulsion

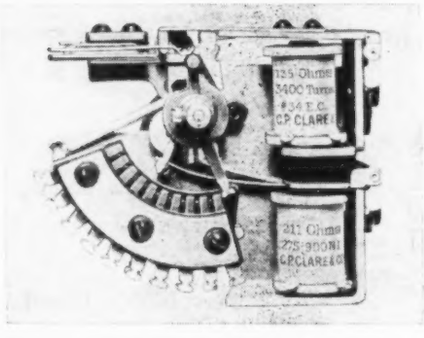
Socony-Vacuum Oil Company, Inc., 26 Broadway, New York 4, N. Y.

Known as S/V Ceremul W, this emulsion is applied by brush. It is diluted in one or two parts of water depending upon the porosity of the surface to which the waterproofing is applied. One application is required. S/V Ceremul W, a dispersion of a blend of petroleum waxes in water, produces a weatherproofing that is claimed not to be re-emulsified by rain, affected by heat or cold, or cracked by vibration.

Multiple-Contact Switch

C. P. Clare & Co., 4719 W. Sunnyside Ave., Chicago 30, Ill.

A ten-point, direct-drive stepping switch, with reset magnet, supplied with one, two or three bank levels. Overall height is 4-1/2 in. width 1-¼ in. and length (from mounting surface) 3-⅝ in. The maximum operating speed of a 48-volt switch under ideal conditions is 36 steps per sec. Release time is 0.03 sec. Operating voltages are—Nominal: 6, 12, 24 and 48; Maximum: 8, 16, 32 and 58. Standard finish of this new Clare switch is cadmium. The net weight ranges from 13 to 19 oz., depending upon the number of bank levels, off-normal springs, and coil windings.



Electronic Relay

Automatic Temperature Control Co., 34 E. Logan St., Phila., 44, Pa.

A relay that utilizes a hot-cathode thyratron. It is snap acting and is claimed to have a high load contact capacity. The unit trips by either an external contact carrying 3 microamperes or by any external circuit the value of which drops to 1 megohm pure resistance. Operation is from 110 v., 60 cy. a.c. lines. Switching action is single-pole double-throw, or to 4-pole double-throw for loads to 12 amp., 110 v., a.c. non-inductive.

This handsome Radio goes to market...

and this metallized trimmer cuts cost!

THAT, in a nutshell, is the story of component parts made of Corning Metallized glass. In this case it's a trimmer capacitor as used in Stromberg-Carlson's beautiful new "Imperial" radio. It eliminates other parts and speeds up assembly.

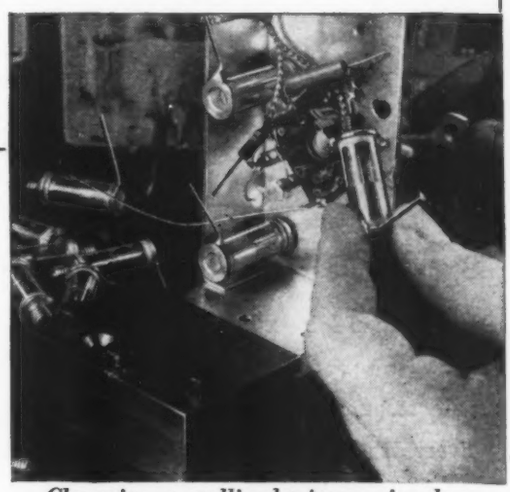
Some other advantages are that it permits permanent, precise and undistorted tuning even under vibration and in high frequency FM bands. Adjustment is made at the electrically cold end of the trimmer. It has a high Q.

It is another example of Corning metallized glass that permits glass parts to be soldered to hardware, permanently, quickly and, where necessary, to form hermetic seals.

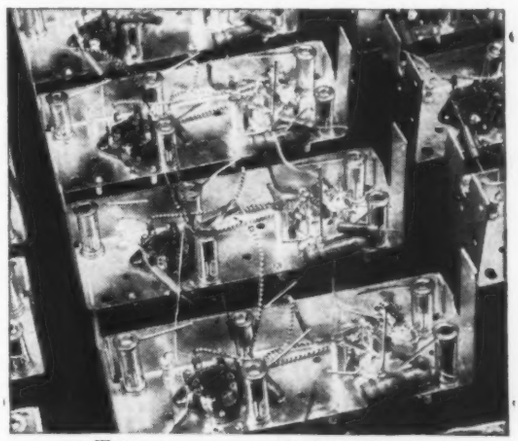
For more information about metallized trimmer capacitors write for Bulletins EI-B-1 and B-2. For the complete story on Corning Metallized Glass Components, write for Bulletin EL-Z-Z-10.

Electronic Sales Department, P-8
Corning Glass Works
Corning, New York

Sales offices in New York, Chicago and San Francisco



Clamping metallized trimmer in place



Trimmers assembled in chassis

CORNING *means* Research in Glass

Electronic Glassware

VYCOR REG. U.S. PAT. OFF. CORNING REG. U.S. PAT. OFF. PYREX REG. U.S. PAT. OFF.

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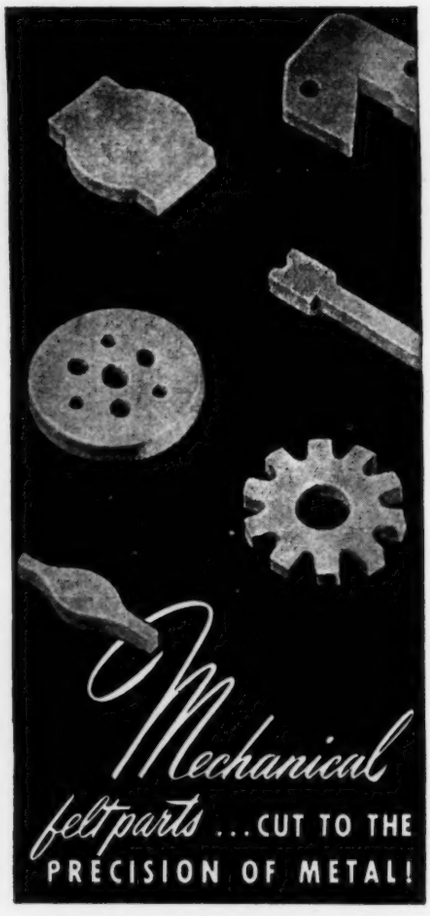

Die-cutting of felt parts is a precision job at Booth . . . often to tolerances usually associated only with metal.

Every order, big or little, is given interested and immediate attention. You receive only precision-cut felts of uniform quality and stamina. We invite your test of Booth Felt Economy.

●

APPLICATION CHART AND SAMPLE KIT ... *contains swatches of S.A.E. felt types, with specification tables. Write for it. (No sales follow-up.)*

THE BOOTH FELT COMPANY
475 19th Street Brooklyn 15, N. Y.
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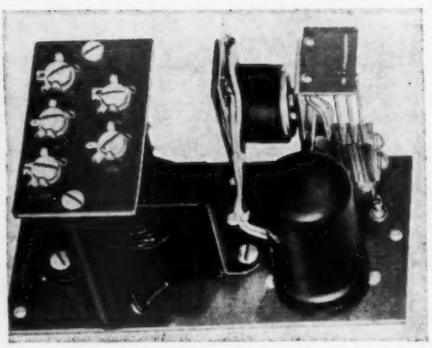
2318

Booth
TRADE MARK
PRECISION CUT FELT PARTS



Mercury contacts are supplied for contacting loads to 30 amp. Up to 100 ft. of conduit, BX or co-axial cable can be used for connections between external actuating source and the relay. Design is suitable for mounting uninclosed.

Drafting Instrument

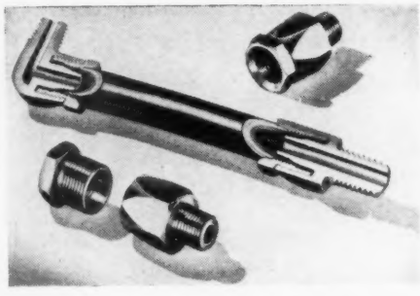
RapiDesign, Inc., P. O. Box 592, Glendale, Calif.

Working on the stencil principle, this device enables the more commonly used standard electric symbols used on electrical circuits and designs to be quickly filled-in. The Electro-Symbol is made from heavy gage, transparent, plastic sheets, with inner contours beveled.

Hose Couplings

Resistoflex Corp., Belleville, N. J.

Two-piece safety-seal couplings for use with any flexible hose. The coupling nut threads onto the hose, with coupling shell compressing the hose end into a safety-seal gasket. The fitting has a long even grip on the hose,



the double bell shape of nut allowing hose to flex without cutting. The body of the fitting prevents crushing or distorting of the line. Large square wrench surfaces are claimed to make assembly operation simple and quick.

Operating Valve

Numatics, Milford, Mich.

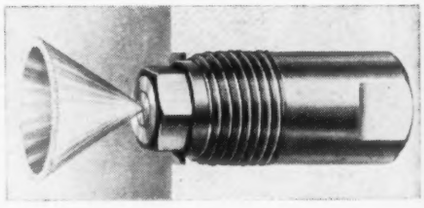
A 4-way hand-controlled unit of the full-ported poppet type. There are no stuffing boxes or springs, all working parts are contained in the upper housing, and sub-bases or mounting brackets are unnecessary. When servicing is required, the housing containing the mechanism can be lifted from the base by removing four socket head screws. The

pipe connections remain undisturbed. These valves, identified as the 11-4 series, are available in six pipe sizes from 1/4 to 1-1/4 in., locking or non-locking.

Atomizing Nozzle

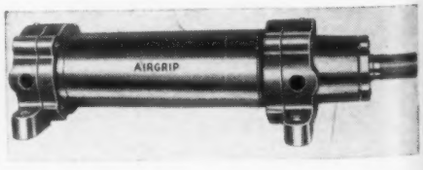
Spraying Systems Co., 4085 West Lake St., Chicago 24, Ill.

An hydraulic unit that can be mounted directly into the wall. The nozzle consists of three parts: (1) Body, (2) cap with stainless steel orifice insert, and (3) body with Monel metal screen or strainer. Water,



oil, solvents, and liquids with similar viscosities can be sprayed. Available in sixteen sizes, these nozzles operate at pressure from 10 to 1,000 lb. per sq. in. Spray is of hollow cone type, with uniform distribution of atomized liquid.

Hydraulic Cylinders

Anker-Holth Mfg. Co., Port Huron, Mich.

Non-rotating hydraulic cylinders having all standard mountings in two pressure ranges having a safety factor of 6 to 1. One range of cylinders is designed for low pressures to 750 lb. per sq.in. the other for high pressures to 2,000 lb. per sq.in. Both are available in sizes from 1½ to 8 in. bore. Any length of stroke can be furnished with standard, oversize, or double-end rod.